GAS AND OIL ENGINES
AND
GAS-PRODUCERS

A TREATISE ON THEORETICAL AND MECHANICAL DEVELOPMENTS OF THE MODERN INTERNAL-COMBUSTION MOTOR, ITS APPLICATION TO THE PRODUCTION OF EFFICIENT POWER UNITS, AND THE LATEST DESIGNS OF FUEL PRODUCERS

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INTRODUCTION

IT is not a mere pleasantry to say that the cannon was the first internal-combustion motor but, although the principle is the same, it is a far cry from the single discharge of burning gas from a big howitzer or the succession of discharges from a machine gun to the orderly and perfectly timed explosions in a single or multi-cylinder internal combustion motor.

¶The difficulties of control, valve action, and timing, which are obvious even to a layman, prevented the rapid development of reliable types during the period from 1860, the date of invention of the Lenoir engine, to the advent of the multi-cylinder automobile motor. In the last decade, however, the best brains of the mechanical world have been developing not only the automobile types with their poppet, sleeve, and rotary valves and their silent and efficient action, but also the larger single cylinder, single- and double-acting types for stationary use, until today the gas engine holds a well recognized place among prime movers.

¶One of the factors which appeals most to the engineer and financier is the high efficiency of the internal combustion motor as compared with the external combustion types like the steam engine and steam turbine and where the question of fuel is easily solved, this argument of efficiency usually carries the day. The development of Diesel types also has had a tendency to broaden the field of activity of the gas engine. This highly efficient motor is capable of burning almost any kind of fuel on account of the high pressure at which ignition takes place and its flexibility has made it particularly adaptable to large units in stationary and marine work.

¶The authors of the work are specialists of the highest order in this line and their professional standing is sufficient guaranty of the scientific accuracy of the article and the practical applications which are given. They have produced a thorough, up-to-date, and reliable treatise, one which will appeal both to the trained engineer and to the layman who is merely interested in this fascinating field of activity.

HORIZONTAL DOUBLE-ACTING GAS ENGINE DRIVING BLOWING ENGINE
Courtesy of The William Tod Company, Youngstown, Ohio

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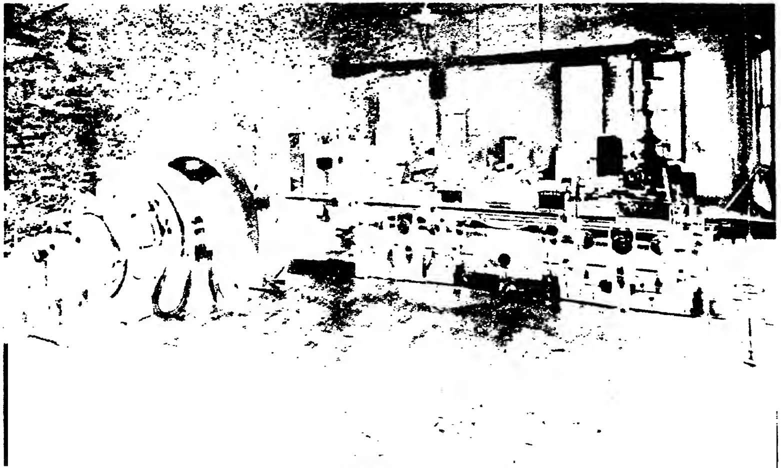


ALLIS-CHALMERS TANDEM GAS ENGINE DRIVING 225 KILOWATT D. C. GENERATOR. INSTALLATION FOR ARMSTRONG CORK COMPANY

Courtesy of Allis-Chalmers Manufacturing Company

PART I

GAS AND OIL ENGINES

INTRODUCTION

Classification of Heat Engines. The heat engine is at present the most important of all the available generators of power. Its purpose is to convert into work the heat derived from the combustion of fuel.

Heat engines may be divided into two broad classes, according to where the combustion of the fuel takes place. In one class the combustion takes place entirely *outside* the working cylinder, the heat of combustion being transmitted by conduction through the walls of a containing vessel to the substance which constitutes the active working agent. Such engines may be called *external-combustion motors*. The most common example of this class is the steam engine. Another example, which is but little used, is the hot-air engine. If the combustion takes place *inside* the engine itself, or in a communicating vessel, so that the products of combustion act directly on the engine, we have an engine of the second class—the so-called *internal-combustion motor*. Gas and oil engines are the most common examples of this type of motor.

EXTERNAL-COMBUSTION MOTORS

Characteristics and Efficiency of Steam Engines. Engines of the second class have certain inherent advantages over external-combustion motors. In the steam engine—practically the most perfect of the external-combustion motors—the heat of combustion generated in the furnace passes through the plates of the boiler to the water on the other side. In the best modern plants, in which the boilers are equipped with superheaters, about 22 to 25 per cent of the heat is wasted during this process by radiation and by loss up the chimney. In plants not equipped with superheaters the water in the boiler is heated to a temperature which does not exceed 400° F., at which temperature its pressure is nearly 250 pounds per

square inch. (If the water were heated to a much higher temperature, the pressure would be too great; for example, at 500° F. the pressure would be 700 pounds, requiring boilers and engines stronger than are at present practicable.) In modern plants, equipped with superheaters, the steam is superheated to a maximum of 650° F., at a pressure of 175 to 200 pounds per square inch. The products of combustion in the furnace have a temperature which is seldom less than 2000° F. Consequently, even with the use of superheated steam, there is necessarily a very large drop in temperature as the heat passes through the boiler plates. The proportion of the total heat going to an engine, which can be converted by the engine into work, depends chiefly upon the temperature range of the working substance; and in the steam engine this range is made comparatively small, not exceeding 300° F. when saturated steam is used, and 550° F. when superheated steam is used.

Consequently, a steam plant not only loses much of its heat up the chimney, but also is able to convert only a small part of the heat that goes to the engine into work. In the best modern steam engines and turbines only about 20 to 22 per cent of the heat going to the engine is converted into work, about 16 to 17 per cent of the heat of combustion of the fuel is converted into work in the best modern steam plants. The ordinary steam engine does not convert into work more than from 6 to 10 per cent of the heat of combustion of the fuel. An economical steam plant consists not only of boilers and engines, but has also a large number of auxiliaries, such as feed pumps, air pumps, condensers, feed-water heaters, economizers, coal conveyors, and steam traps. After shutting down, it requires considerable time and fuel to raise steam in the boilers before the plant can be put again in operation; or, if the fires are kept banked so as to maintain steam pressure while the engines are not running, a considerable amount of fuel will be used for this purpose without any corresponding work being done.

INTERNAL-COMBUSTION MOTORS

Characteristics and Efficiency of Gas Engines. In the internal-combustion motor, where the fuel is a gas or volatile oil, there is no apparatus corresponding to a boiler, and no losses corresponding to the boiler losses. If the fuel is coal, it has to be converted into gas

before it can be used in an internal-combustion motor; and this necessitates the use of a *gas producer*, in which some heat will be lost, though not so much as is usual in a boiler. The fuel, being burned in the engine, gives there a temperature of from 2000° F. to 3000° F., so that the temperature range in the engine is very large—from 2 to 3 times that obtaining in a steam engine; consequently, the engine can be more efficient—that is, can convert a larger proportion of the heat of combustion into work—than in a steam plant. The high temperatures are not necessarily accompanied by high pressures, because it is air—not hot water—which is heated to those temperatures. In practice, the best internal-combustion motors have converted 35 per cent of the heat of combustion into work, or twice as much as the best steam engines; and the ordinary small gas engine will convert from 15 to 20 per cent of the heat of combustion into work. The internal-combustion plant is also much simpler, having but few auxiliaries. The number of men necessary to run a large gas-engine plant is small. The plant is ready to start up at a minute's notice, and the standing losses are very small or nothing. When a liquid fuel is used, the absence of a boiler or other auxiliaries makes the internal-combustion motor lighter, more compact, and more easily portable than any other motor. The absence of a boiler also does away with the risk of disastrous explosions. Consequently, no inspection is required by law, no license is necessary for running the plant, and lower rates for insurance are secured.

The practical use of the internal-combustion motor is a comparatively recent development. The last twenty years have brought about great improvements in its operation, a marked increase in its use, and a large extension in its various applications. The internal-combustion motor is less uniform in its speed of rotation, and is more liable to derangement than the steam engine; but these difficulties have been largely overcome, so that modern gas engines are used for electric lighting, and have a reliability but little short of that of the steam engine.

Fuels Used. The fuels used in external-combustion motors may be solid, liquid, or gaseous. In internal-combustion motors, solid fuels must be gasified before they are taken into the engine, because the incombustible matter, or ash, present in them, would rapidly destroy the rubbing surfaces in the cylinders. The actual

fuels going to the engine are either gaseous or liquid, and the latter may be sent into the cylinder either as a vapor or as a liquid. There is no essential difference between engines using gas and those using oil; the cycle of operations occurring in the cylinder is the same with both kinds of fuel; the only differences are structural and the addition of special apparatus for vaporizing the oil. An engine can be, and often is, quickly converted from a gas to an oil engine.

In the present work, whatever presentation is made of thermodynamic theory applies to both gas and oil engines. The features of oil engines are treated after the discussion of the gas engine.

HISTORICAL SKETCH

The history of the development of the internal-combustion motor begins with the invention of cannon. A gun is a motor in which the working substance is the gas resulting from the combustion of the powder, and in which work is done on the projectile, giving it kinetic energy. Such a motor is not continuous in its action, but it offers possibilities of a practicable engine if the powder charge is small and the projectile or piston on which the gases act is restricted in its movement. The earliest internal-combustion motors devised for doing useful work were intended to use gunpowder. The first of these was suggested by Abbé Hautefeuille in 1678, and was followed shortly by others, none of which were practically realizable in the state of the mechanic arts at that time.

It was not until the discovery by Murdock, near the end of the eighteenth century, that a combustible gas could be obtained from coal by a process of distillation, that a practical internal-combustion motor was possible. As soon as the properties and method of manufacture of coal gas became known, numerous attempts were made to use it in engines. Up to the year 1860, many engines were devised and patented, and in several cases constructed, operated, and sold. None of these engines, however, can be said to have been satisfactory. They were irregular in action, noisy, wasteful of fuel, and in general had practical defects.

Lenoir Engine. The Lenoir engine, which appeared in 1860, was the first really practical gas engine. Hundreds of these engines were made and sold; and the greatest interest in this type was aroused in France, where it was built, and in England, where it was largely used.

In general appearance, the engine resembles a double-acting horizontal steam engine. The cylinder, shown in horizontal section in Fig. 1, has a separate admission port *a* and exhaust port *b* at each end. The valves are simple slide valves, driven by eccentrics, and so designed that the inside edges alone uncover the ports. The valve *G* is used for the admission of the explosive mixture, which consists of air entering the valve cavity from *d* and gas coming through one of the branches *r* of the gas pipe, and passing through the hole *i* in the valve. The air and gas enter the port *a* through a number

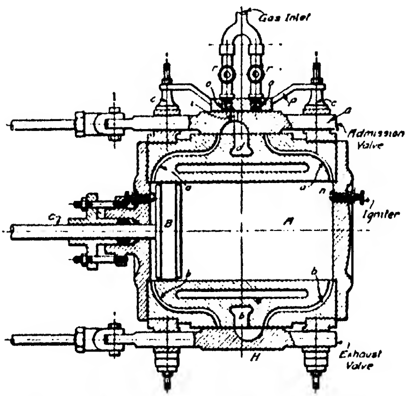


Fig. 1 Horizontal Section of Cylinder and Valves of Lenoir Engine

of small holes, in which they are thoroughly mixed; and the mixture is exploded in the cylinder, when desired, by an electric igniter *n*. The exhaust is through the port *b* and the cavity in the exhaust valve *H* to the atmosphere. As the cylinder rapidly becomes very hot, it is provided with a water jacket.

The series or cycle of operations which takes place in this engine is as follows: During the first part of the stroke, the admission valve *G* uncovers the port *a*, so that a mixture of air and gas enters the cylinder, filling the space behind the piston. At half-stroke, the

valve closes the port, and a spark from an induction coil passes between the terminals n of the electric igniter, exploding the mixture and raising its pressure to 60 or 70 pounds per square inch. The piston is then forced to the end of its stroke, the products of combustion expanding behind it. At the end of the stroke, the valve H uncovers the exhaust port, and keeps it open throughout the whole of the return stroke, so that all the products of combustion are expelled to the atmosphere. A similar cycle of operations occurs on the other side of the piston. In Fig. 1, the valve G is just opening the port at the left, so that admission may take place there; and the valve H is just opening the port at the right, so that exhaust may occur from the other end of the cylinder. A reproduction of an indicator card from this engine is shown in Fig. 2.

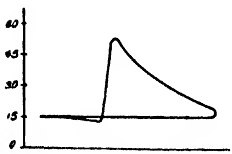


Fig. 2. Indicator Card of Lenoir Engine

This engine gave considerable trouble in many cases, but the principal reason for the falling-off in its use was the large amount of gas it required. It used from 60 to 70 cubic feet of coal gas per indicated horsepower per hour, or from three to four times as much as a modern gas engine, so that it did not compare very favorably with the steam engine in its running cost.

Otto Engine. *Four-Stroke Otto Cycle.* In the year 1862, it was pointed out by a French engineer, Beau de Rochas, that in order to get high economy in a gas engine certain conditions of operation were necessary. The most important of these conditions is that the explosive mixture shall be compressed to a high pressure before ignition. In order to accomplish this, he proposed that the cycle of operations should occupy four strokes, or two complete revolutions, of the engine, and that the operations should be as follows:

(1) *Suction* or *admission* of the charge of gas and air throughout the complete forward stroke.

(2) *Compression* of the explosive mixture during the whole of the return stroke, so that it finally occupies only the clearance space.

(3) *Ignition* of the charge at the end of the second stroke, and *expansion* of the exploded mixture throughout the whole of the next forward stroke.

(4) *Exhaust*, beginning at the end of the forward stroke, and continuing throughout the whole of the last return stroke.

Fig. 3 is a diagram showing the operations of the four-stroke cycle.

This cycle was not actually used until 1876, when Dr. Otto adopted it in his engine and thereby produced the modern gas engine. The four-stroke cycle of Beau de Rochas is now universally known as the "Otto Cycle".

Diesel Engine. *Four-Stroke Diesel or Constant-Pressure Combustion Cycle.* In 1872, Brayton took out American patents on an engine adapted to burn gas, and in 1874 patents covering an engine adapted to burn liquid hydrocarbons. These engines were the first in which the fuel was burned at constant pressure. It was a thoroughly practical machine and found considerable application, but its maximum pressure was only about 45 pounds per square inch; consequently, its econom was so low as to prohibit its competing with other engines.

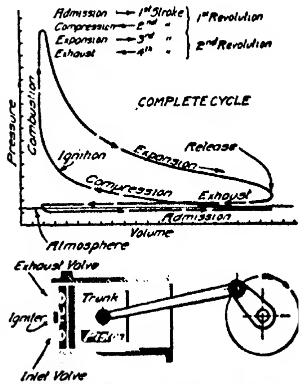


Fig. 3. Diagram Showing Operation of Otto Four-Strok Cycle; Lower Part of Diagram, Near Atmosphere, Exaggerated

In 1895, Dr. Rudolph Diesel produced an engine which, with i improvements and modifications, is, to date, the most importa development in heavy oil engines. In this engine, air alone is con pressed in the engine cylinder to such a pressure as to heat it abo the ignition point of the oil fuel. As this high pressure is reach gradually, it does not cause a shock to the engine, such as a explosion which reached the same pressure would give.

At the end of the compression stroke, when the air is at this hig pressure and temperature, the oil is injected into the cylinder by a

which has been compressed to a still higher pressure in a separate, small compressor. The oil spray is vaporized, or atomized, and is ignited as it enters, by the hot air. Since the piston is moving forward, and the oil vapor is burned as it enters, the combustion can be made to occur without increase of pressure; in the actual engine the pressure drops slightly. The method of burning is, in fact, essentially similar to that of an ordinary gas burner, and not to that of an explosive mixture; consequently the oil will burn with any excess of air present. The power of the engine is regulated by governing the proportion of the stroke during which oil is injected. The other events in this cycle take place exactly as in the Otto

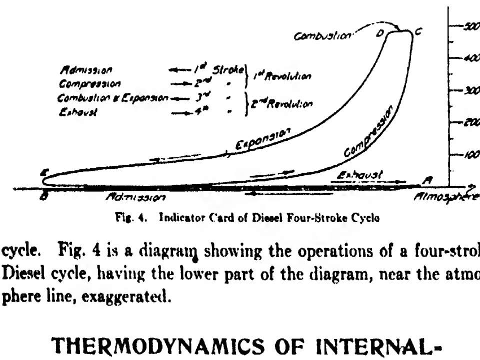


Fig. 4. Indicator Card of Diesel Four-Stroke Cycle

cycle. Fig. 4 is a diagram showing the operations of a four-stroke Diesel cycle, having the lower part of the diagram, near the atmosphere line, exaggerated.

THERMODYNAMICS OF INTERNAL-COMBUSTION CYCLES

OTTO CYCLE

Explosive Mixture. In Otto-cycle engines the explosive mixture in the cylinder consists of air mixed with a smaller volume of the gaseous or liquid fuel. For instance, if the engine uses gas supplied from the city mains, the mixture will average about 8 or 9 parts of air to 1 of gas, and should never have less than about 6 parts of air to 1 of gas. This mixture will behave, up to the time when the explosion takes place, as if it were pure air. Also, the products of combustion, after the explosion is completed, have physical proper-

ties only slightly different from those of air; and, consequently the working substance in the cylinder can be regarded, without serious error, as consisting entirely of air. In the following discussion of what occurs in the engine cylinder, it is assumed throughout that the substance in the cylinder has the physical properties of air.

Ideal Cycle. *Admission Stroke.* The processes taking place in the cylinder of an ideal engine are best represented on a pressure-volume diagram. At the beginning of the cycle the piston is at the end of its path and is about to begin its outstroke. The clearance space is full of products of combustion; the pressure is atmospheric pressure, because the cylinder has been in communication with the atmosphere through the exhaust valve, which has just closed. The condition existing in the cylinder at this instant is represented in the diagram, Fig. 5, by the point *1*, which is at a horizontal distance from the vertical axis representing the clearance volume, and at a vertical distance above the horizontal axis representing the atmospheric pressure of 14.7 pounds per square inch. As the piston makes its outstroke, the admission valve opens, admitting the charge to the cylinder throughout the stroke; and, as the cylinder is in communication with the outside air through the air-admission valve, the pressure in the cylinder remains atmospheric pressure throughout the stroke. On the diagram the admission is represented by the line *1–2*, which is at the constant height, representing the atmospheric pressure, and whose length represents the volume of the charge taken in, which is the same as the volume through which the piston moves. The point *2* represents the condition at the end of the first stroke.

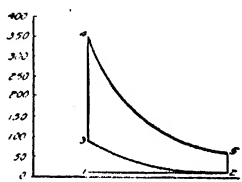


Fig. 5. Ideal Indicator Card of Otto Cycle

Compression Stroke. The admission valve now closes, and the piston makes its return stroke. Since all the valves are closed, the charge cannot escape and is crowded into a smaller volume, while its pressure rises. The process continues until the piston reaches the end of its stroke, at which time the whole charge is compressed into the clearance space. This process is represented by the line *2–3*, which shows the rise in pressure resulting from the compression.

A compression of this kind, occurring without the addition or the abstraction of heat from the gas, is called an *adiabatic compression*. It causes an increase, not only in the pressure, but also in the temperature of the gas. It is the process which takes place in the working of an ordinary bicycle pump, and which causes its rise in temperature. The relation between the pressure of air and its volume when subject to adiabatic compression is

$$PV^{1.405} = \text{constant}$$

Note carefully that in this equation P means the absolute pressure, and not the pressure shown by a gage. If exchanges of heat occur between the gas and the cylinder while the compression is taking place, the relation between the pressure and the volume of the air can usually be represented by the equation

$$PV^{n} = \text{constant}$$

where n has a value greater than 1.405 if heat is added to the gas, and less than 1.405 if heat is being abstracted from it, during the compression. Compressions following this equation are often called "polytropic compressions".

Power Stroke. When the charge has reached the condition represented by the point *3*, it is ignited, and the heat generated by the explosion raises the temperature, and consequently the pressure, of the mixture. The combustion occurs so rapidly that the piston has not time to start on the outstroke before the combustion is completed, and the rise of pressure occurs, as is shown by the line *3–4*, while the volume of the gas is constant. The hot products of combustion at the pressure P_4 now force the piston out, and, expanding behind it, they fall in pressure. This expansion, occurring without communication of heat to or from the gas, is *adiabatic expansion*, and is consequently accompanied by a fall in temperature of the gas. If heat is added or abstracted, the expansion is *polytropic*, but not adiabatic. The equations of adiabatic or polytropic expansion curves are the same as those of similar compression curves.

Exhaust Stroke. At the point *5* the piston is at the end of the stroke, and no more expansion is possible. The exhaust valve opens, and the pressure in the cylinder falls immediately to atmospheric pressure, as shown by the line *5–2* in the diagram. Throughout the last return stroke, *2–1*, the exhaust valve remains open, so

that the pressure in the cylinder remains atmospheric pressure. The completed diagram, Fig. 5, shows the whole series of pressure and volume changes occurring in a gas engine, and is such a diagram as would be taken by an indicator from a perfect engine. The area *2-3-4-5* enclosed by the diagram represents the work done by the engine per cycle.

Pressures and Temperatures During the Cycle. The pressures and temperatures of the working substance, and the amount of work

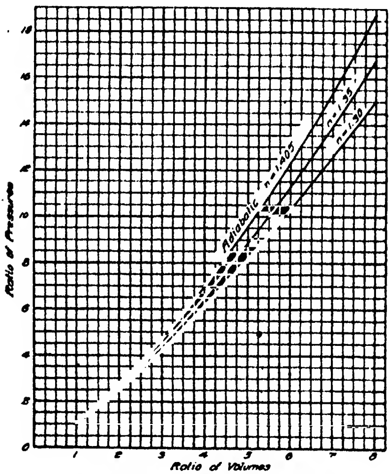


Fig. 6. Polytropic Expansion or Compression Curves for Otto Cycle

done in an engine which exactly follows the Otto cycle, can be readily calculated. Starting at the point *2*, Fig. 5, there is present in the cylinder a volume V_2 at atmospheric pressure P_2 and at the temperature t_2, which will be assumed to be the temperature of the air as it came into the cylinder.

Pressure After Compression of Gas. The working substance is compressed adiabatically until it fills only the clearance volume V_3.

The consequent rise in pressure can be calculated from the formula already given; but it is more simply obtained from the adiabatic curve, Fig. 6, which gives the relation between the changes of volume and of pressure in adiabatic expansion or compression. The horizontal scale in this diagram is the ratio of expansion or compression, and the vertical scale shows the corresponding ratio of the pressures at the beginning and end of the expansion or compression. If, for example, the working substance expands adiabatically to five times its original volume, the pressure (which varies inversely as the volume), is shown by the curve to fall to $\frac{1}{9.67}$ of its original value. Conversely, if the working substance is compressed to $\frac{1}{5}$ the original volume, the pressure rises to 9.67 times its original value. Consequently, the pressure at the point *3*, Fig. 5, can be found by the use of this curve.

Example. A gas engine with $33\frac{1}{3}$ per cent clearance takes in its charge at 14.7 pounds per square inch pressure. What is the pressure at the end of the adiabatic compression?

Solution. The clearance volume V_3 is $33\frac{1}{3}$ per cent of the volume V_2-V_3, through which the piston moves; or

$$V_3=\frac{33\frac{1}{3}}{100}(V_2-V_3)$$
$$=\tfrac{1}{3}(V_2-V_3)$$
$$3V_3=V_2-V_3$$
$$4V_3=V_2$$

and

$$\frac{V_2}{V_3}=4$$

From the curve, Fig. 6, if the ratio of compression is 4, the corresponding ratio of pressures is 7 (a more accurate determination shows this value to be 7.06, which will be used in the subsequent problem), so that the pressure at the end of compression is 7.06 times the pressure at the beginning of compression. Therefore, the pressure at the end of compression, $P_3=7.06\times14.7$, or 103.8 pounds per square inch, absolute.

Temperature After Compression of Gas. The temperature at the end of the adiabatic compression, or other polytropic compressions, can be found from the equation for a perfect gas. This may be stated in the form

$$PV=wRT$$

where w is the weight of the gas; R is a constant for any perfect gas, and has the value 53.2 for air; P is the pressure in pounds per square foot, absolute; and T is the absolute temperature of the gas. The weight of the gas is constant throughout the adiabatic compression, and can be found from the point *2* if P_2, V_2, and T_2 are known. The temperature at *3* can then be found from the equation

$$P_3V_3 = wRT_3$$

where P_3 denotes the pressure in pounds per square foot. Hence, to reduce pressures expressed in pounds per square inch to pressures per square foot, a factor 144 must be used as a multiplier.

Example. Assuming the conditions of the previous problem, and supposing the temperature of the air to be 60° F., what is the temperature of the charge at the end of the compression?

Solution.

$$P_2V_2 = wRT_2$$

$$\therefore \qquad wR = \frac{P_2V_2}{T_2} = \frac{14.7 \times 144 \times V_2}{60 + 461}$$

Also

$$T_3 = \frac{P_3V_3}{wR}$$

Rearranging terms and substituting for wR its value from the preceding equation, we have

$$T_3 = P_3 \frac{V_3}{V_2} \times \frac{60 + 461}{14.7 \times 144}$$

but

$$\frac{V_3}{V_2} = \frac{1}{4}$$

Therefore, substituting the values of P_3 and $\frac{V_3}{V_2}$, we have

$$T_3 = 103.8 \times 144 \times \frac{1}{4} \times \frac{521}{14.7 \times 144} = 919.6° \text{ absolute}$$

and

$$t_3 = 458.6° \text{ F}$$

Temperature Rise During Explosion. The rise in temperature during explosion depends on how much heat is generated, which, in turn, depends on the strength of the explosive mixture and the heat of combustion of a cubic foot of the fuel. Let H be the heat of combustion of a cubic foot of the fuel in B.t.u., and let the mixture consist of 1 part of gas to L parts of air. The total volume of the charge taken into the cylinder each admission is

$$V_2 - V_1 \text{ cu. ft.}$$

the volume of fuel in this charge

$$\frac{1}{L+1}(V_2-V_1)$$

and the heat of combustion Q of this fuel is

$$Q=H\,\frac{V_2-V_1}{L+1}\ \text{B.t.u.}$$

This heat is utilized in raising the temperature of the gas from the known temperature T_3 to another temperature T_4. The rise in temperature can be found when the heat necessary to raise one pound of air one degree in temperature is known. This amount of heat is called the *specific heat.* It is represented by the symbol C_v (the suffix indicating that the volume is unchanged while the temperature rises), and is equal to .169 B.t.u. for air. With a weight of w pounds, the heat necessary to raise the gas one degree in temperature is

$$wC_v\ \text{B.t.u.}$$

To raise the temperature T_4-T_3 degrees, the heat supply is

$$wC_v(T_4-T_3)\ \text{B.t.u.}$$

and the heat of combustion is used entirely in raising the gas from T_3 to T_4.

Therefore

$$H\frac{V_2-V_1}{L+1}=wC_v(T_4-T_3)$$

Substituting for w its value from the general equation

$$H\frac{V_2-V_1}{L+1}=\frac{P_2V_2}{RT_2}C_v(T_4-T_3)$$

Rearranging terms and solving for T_4-T_3, we have

$$T_4-T_3=\frac{H}{L+1}\times\frac{RT_2}{P_2}\times\frac{1}{C_v}\times\frac{V_2-V_1}{V_2}$$

Example. In the previous problem, if the charge taken in consists of 1 part of gas to 7 parts of air, and the heat of combustion of the gas is 640 B.t.u. per cubic foot, find the temperature at the end of explosion.

Solution.

$R=53.2$; $T_2=521$; $H=640$; $P_2=14.7$; $C_v=.169$; $L+1=7+1=8$; and $T_3=919.6$

$V_2-V_1=V_2-V_3$; because volume $V_1=V_3$ according to Fig. 5.

$$\frac{V_3}{V_2}=\frac{1}{4}\text{ and }\therefore\frac{V_1}{V_2}=\frac{1}{4}\text{ or }V_1=\frac{1}{4}V_2$$

$$\frac{V_2-V_1}{V_2}=\frac{V_2-\frac{1}{4}V_2}{V_2}=\frac{\frac{3}{4}V_2}{V_2}=\frac{3}{4}$$

Substituting the proper values in the equation, we have

$$T_4 - T_3 = \frac{640}{8} \times \frac{53.2 \times 521}{14.7 \times 144} \times \frac{1}{169} \times \frac{3}{4}$$

$$T_4 = 4649 + T_3$$
$$= 5568.6° \text{ absolute}$$
$$t_4 = 5107.6° \text{ F}$$

If a perfect gas is raised in temperature, while its volume is unchanged, the absolute pressure will increase in exact proportion to the rise of absolute temperature; or

$$P_4 : P_3 : : T_4 : T_3$$

$$\therefore \quad P_4 = \frac{T_4}{T_3} P_3$$

Example. What is the pressure at the end of explosion in the preceding problem?

Solution.

$$P_4 = \frac{T_4}{T_3} P_3$$
$$= \frac{5568.6}{919.6} \times 103.8 \text{ lb. per sq. in., absolute}$$
$$= 628.6 \text{ lb. per sq. in., absolute}$$

Pressure and Temperature at End of Expansion. The pressure and temperature at the end of the adiabatic expansion can be found most simply, after the other pressures and temperatures are known, by making use of a relation which exists between the pressures and temperatures at the points *2*, *3*, *4*, *5* *. These relations are

$$\frac{P_2}{P_3} = \frac{P_5}{P_4} \text{ and } \frac{T_2}{T_3} = \frac{T_5}{T_4}$$

Example. What are (*a*) the pressures, and (*b*) the temperatures, at the end of the adiabatic expansion in the preceding problem?

Solution.

(*a*) $P_5 = \frac{P_2}{P_3} \times P_4 = \frac{14.7}{103.8} \times 628.6 = 89$ lb. per sq. in., absolute

(*b*) $T_5 = \frac{T_2}{T_3} \times T_4 = \frac{521}{919.6} \times 5568.6 = 3155°$ absolute $= 2694°$ F.

Work Done by Heat Engine. The work done by any heat engine is equal to the difference between the heat which goes to the engine and that which is rejected by the engine, because whatever heat disappears cannot have been destroyed and must have been

*The ratio of the pressures $\frac{P_4}{P_5}$ can be obtained from the curve, Fig. 6, since the ratio of the volumes $\frac{V_5}{V_4}$ is known. But $V_5 = V_2$; therefore $\frac{V_5}{V_4} = \frac{V_2}{V_3}$ and $\frac{P_2}{P_3} = \frac{P_5}{P_4}$.

converted into work. In the Otto cycle the heat taken in has been seen to be

$$Q = wC_v(T_4 - T_3) \text{ B.t.u.}$$

Heat is rejected from the engine only during the process represented by the line *5–2*, because, when the charge gets back to the condition *2*, it has returned to its original volume and pressure, and consequently to its original temperature. The heat rejected is then:

$$Q_R = wC_v(T_5 - T_2) \text{ B.t.u.}$$

And consequently, the work done per cycle is found by subtracting the rejected heat from the heat taken in and substituting for the B.t.u. its equivalent, 778 ft.-lb. Thus:

$$W = (Q - Q_R) \text{ B.t.u.} = 778\ (Q - Q_R) \text{ ft.-lb.}$$

Efficiency of Cycle The efficiency of the cycle—that is, the fraction of the heat supplied that is converted into work—is

$$E = \frac{W}{Q} = \frac{Q - Q_R}{Q}$$

$$= 1 - \frac{Q_R}{Q}$$

$$= 1 - \frac{wC_v(T_5 - T_2)}{wC_v(T_4 - T_3)}$$

$$= 1 - \frac{T_5 - T_2}{T_4 - T_3}$$

And since, as already stated $\frac{T_5}{T_4} = \frac{T_2}{T_3}$

we get by substitution:

$$\frac{T_5 - T_2}{T_4 - T_3} = \frac{T_2}{T_3}$$

therefore

$$E = 1 - \frac{T_2}{T_3}$$

Example. Find the efficiency of the cycle in the preceding problem.
Solution.

$$E = 1 - \frac{T_2}{T_3}$$

$$= 1 - \frac{521}{919.6}$$

$$= 1 - .567 = .433$$

Horsepower Calculations. The work W done per cycle can be calculated from the efficiency without knowing the heat rejected.

$$E=\frac{W}{Q}$$

or

$$W=E\times Q \text{ B.t.u.}$$
$$=778E\times Q \text{ ft.-lb.}$$

Example. If the cycle discussed in the previous examples takes place in a cylinder of 12 inches diameter and 18 inches stroke, what will be the work done per cycle? If the engine makes 250 revolutions per minute, what will be its indicated horsepower.

Solution.

$$W=778E\times Q \text{ ft.-lb.}$$
$$Q=\frac{H}{L+1}(V_2-V_1) \text{ B.t.u.}$$

V_2-V_1 is the volume (in cubic feet) through which the piston moves, and is the product of the cross-sectional area of the cylinder in square feet by the stroke in feet.

$\therefore$ $$V_2-V_1=\frac{\pi}{4}\times\left(\frac{12}{12}\right)^2\times\frac{18}{12}=1.178 \text{ cu. ft.}$$

and

$$Q=\frac{H}{L+1}(V_2-V_1)$$

But

$$H=640;\ L+1=8;\ \text{and } V_2-V_1=1.178$$

$\therefore$ $$Q=\frac{640\times 1.178}{8}=94.24 \text{ B.t.u.}$$

also

$$W=E\times Q$$

and if

$$E=.433, \text{ and } Q=94.24$$

$\therefore$ $$W=.433\times 94.24=40.81 \text{ B.t.u.}$$
$$=31{,}750 \text{ ft.-lb.}$$

Since this engine requires two revolutions to complete a cycle, the number of cycles per minute is only half the number of revolutions per minute, therefore

$$\text{Work per minute}=W\times 125 \text{ ft.-lb.}$$

and

$$\text{Horsepower}=\frac{31{,}750\times 125}{33{,}000}=120.3 \text{ i.h.p.}$$

EXAMPLES FOR PRACTICE

A gas engine using the Otto cycle has 25 per cent clearance, and takes in its charge at 14.7 pounds per square inch and at 60° F.

1. What is the pressure at the end of the compression?

Ans. 141.1 lb. per sq. in., absolute

2. What is the temperature at the end of compression?

Ans. 539° F.

3. If the charge consists of 1 part of gas to 9 parts of air, and the heat of combustion of the gas is 600 B.t.u. per cubic foot, what is the temperature at the end of explosion? Ans. 4,258° F.

4. What is the pressure at the end of explosion?

Ans. 665.9 lb. per sq. in., absolute

5. What are the pressure and temperature at the end of the expansion? Ans. 69.4 lb. per sq. in., absolute. 1,997° F.

6. What is the efficiency of the cycle? Ans. .479

7. If the cylinder diameter is 18 inches, the stroke 24 inches, and the engine makes 150 revolutions per minute, what is the i.h.p.?

Ans. 180 i.h.p.

Changes in Calculations for Polytropic Reactions. While in the ideal engine the compression and expansion are adiabatic (with the exponent n equal 1.405), in the real engine the exponent n may vary between 1.30 and 1.35. Recalculating the examples on pages 12 to 17, using n equals 1.30 in place of n equals 1.405, gives the following results:

If the working substance expands with the exponent n equals 1.30 to five times its original volume, the pressure is shown by Fig. 6 to fall to $\frac{1}{8.12}$ of its original value; and conversely if the working substance is compressed.

Example. Same problem as in the middle of page 12. What is the pressure at the end of the compression when $n = 1.30$?

Solution. $\frac{V_1}{V_2} = 4.$ From the curves, Fig. 6, if the ratio of compression is 4, the corresponding ratio of pressure is 6.08. Therefore, the pressure at the end of compression is

$$P_2 = 6.08 \times 14.7 \text{ or } 89.4 \text{ lb. per sq. in., absolute}$$

Example. Same as page 13, except that, instead of adiabatic compression, $n = 1.30$.

Solution.

$$T_2 = 89.4 \times 144 \times \tfrac{1}{4} \times \frac{521}{14.7 \times 144} = 792.0° \text{ absolute}$$

$$t_2 = 331.0° \text{ F.}$$

Example. Same as bottom of page 14.

Solution.

$$T_3 = 4649 + 792.0 = 5441.0° \text{ absolute}$$
$$t_3 = 4980.0° \text{ F.}$$

Example. Same as in middle of page 15.
Solution.

$$P_4=\left(\frac{T_4}{T_3}\right)\times P_3=\frac{5441\times 89.4}{792}=614.1 \text{ lb. per sq. in., absolute}$$

Example. Same as bottom of page 15.
Solution.

$$(a)\quad P_1=\left(\frac{P_2}{P_3}\right)\times P_4=\frac{14.7\times 614.1}{89.4}=101.0 \text{ lb. per sq. in., absolute}$$

$$(b)\quad T_1=\left(\frac{T_2}{T_3}\right)\times T_4$$

$$=\frac{521\times 5441}{792}=3579° \text{ absolute}=3118° \text{ F.}$$

Example. Same as bottom of page 16.
Solution.

$$E=1-\left(\frac{T_2}{T_3}\right)=1-\frac{521}{792.0}=1-0.658=0.342$$

Example. Same as in middle of page 17.
Solution.

$$(a)\quad W=94.24\times 778\times 0.342=25{,}075 \text{ ft.-lb.}$$

$$\text{h.p.}=\frac{(25{,}075\times 125)}{33{,}000}=95.0 \text{ i.h.p.}$$

From a comparison of these results with those where the expansion and compression are assumed to be adiabatic, an idea of the effect of a variation from ideal conditions can be gained.

EXAMPLES FOR PRACTICE

Recompute the examples, 1 to 7, on pages 17 and 18 on the assumption that, instead of adiabatic compression and expansion, the compression and expansion are polytropic, with an exponent of (*a*) *n* equals 1.35, and (*b*) *n* equals 1.30.

Ans.:

1. (*a*) 129.1 lb. per sq. in., absolute
   (*b*) 119.7 lb. per sq. in., absolute
2. (*a*) 454° F
   (*b*) 383.1° F.
3. (*a*) 4,173° F.
   (*b*) 4,102° F.
4. (*a*) 653.9 lb. per sq. in., absolute
   (*b*) 644.1 lb. per sq. in., absolute
5. (*a*) 74.5 lb. per sq. in., absolute
   2178° F
   (*b*) 79.5 lb. per sq. in., absolute
   2355° F.
6. (*a*) 0.431
   (*b*) 0.383
7. (*a*) 161.6 i.h.p.
   (*b*) 143.6 i.h.p.

Otto Cycle with Increased Expansion. The pressure at the end of expansion is seen in the example on page 15 to be 89 pounds per square inch, absolute. In ordinary practice it is commonly found to be from 50 to 60 pounds, absolute. It is evident that if the gas were permitted to expand further, it would do more work, and, consequently, would increase the efficiency of the cycle. The indicator card, Fig. 7, shows one method used for obtaining more expansion. The charge enters at atmospheric pressure from *1* to *2*, when the admission is cut off. The piston continues moving forward to the end of its stroke, but as no more admission takes place the charge expands adiabatically to *3*, while its pressure falls. On the return stroke the charge is compressed adiabatically, retracing the expansion path along *3*-*2*, and continuing till the whole charge is compressed into the clearance space at *4*. The rest of the cycle is unchanged. Diagram *1*-*2*-*4*-*5*-*8*-*2* represents the ordinary Otto cycle, and the shaded area *8*-*6*-*7*-*2* represents the increase in work due to the increased expansion.

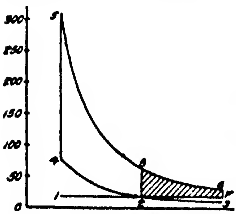


Fig. 7. Method of Increasing Expansion

Efficiency Dependent upon Clearance. An examination of the equation for the efficiency of the Otto cycle

$$E = 1 - \frac{T_1}{T_2}$$

brings out certain important results. The efficiency is seen to depend only on the ratio of the temperatures at the beginning and end of compression, and not at all upon the temperature and pressure at the end of explosion. Since the ratio of the temperatures at the beginning and end of compression depends only upon the ratio of compression, and since, further, the charge is always compressed until it occupies the clearance volume, the efficiency is seen to depend only upon the percentage clearance. In other words, in engines with the same percentage clearance, using the Otto cycle, the percentage of the heat liberated in the cylinder which is converted into work is always the same, whatever be the size of the engine or the

TABLE I

Effects of Clearance

Percentage Clearance of Otto Cycle Engine	Pressure at End of Compression Lbs. per Sq. In.	Efficiency of Otto Cycle	Efficiency of Cycle with Increased Expansion, but with Same Compression Pressure as Otto Cycle
20	183.3	51.6	60.9
25	141.1	47.9	58.4
30	115.4	44.8	55.0
35	98.0	42.1	52.5
40	85.5	39.8	50.4

strength of the charge. The effect of the clearance on the efficiency is exhibited in Table I, where it is seen that the smaller the clearance the greater is the efficiency of the engine. The pressures at the end of compression are also given in the table, and are calculated on the assumption that the atmospheric pressure is 14.7 pounds per square inch, absolute. The efficiency of this cycle, with increased expansion, can be easily calculated, and the results of such calculations are given in Table I. They are made on the assumption that the charge is admitted for only one-half the stroke, and that the heat of combustion is 80 B.t.u. per cubic foot of the charge. An inspection of Table I shows the increase in efficiency which results from the increased expansion for engines which have the same pressures at the end of compression, and indicates that, to be of high efficiency, a gas engine of this type should first compress the charge to a high pressure, and then expand the products of combustion to a volume considerably in excess of the original volume of the charge.

Ideal and Real Otto Cycles. The calculations in the preceding pages are made on the assumption that the gas engine follows the Otto cycle exactly, in which case the engine is called an *ideal* engine. The *real* engine does not exactly follow the Otto cycle, because of certain practical difficulties. The actual indicated work is always less than the theoretical in consequence of incomplete combustion, losses due to cooling and radiation, etc.

Departures from Ideal Conditions at Each Stroke. Differences between the real and the ideal engine occur in each part of the cycle. During admission, Fig. 8, line *1–2*, the pressure in the cylinder is

TABLE II

Table of the Values of the Compression Pressures (p_c)—Compression Temperatures (T_c° F. Absolute)— and the Theoretical Efficiency (E)—for Different Values of the Percentage Clearance (c)—and Polytropic Exponent (n) of the Compression Line

(The pressure during the suction stroke (p_s) is taken as 12.5 lb. per sq. in. absolute,* and the temperature of the mixture at the end of the suction stroke is taken as $T_s = 700°$ F. absolute (t_s equals 240° F.)

Percentage Clearance c =		40	33.3	28.6	25	20	16.6	14.3	12.5	11.1
n = 1.30	p_c =	63.7	75.8	88.3	101.3	128.4	156.9	186.6	217.5	249.4
	E =	0.313	0.340	0.363	0.383	0.416	0.442	0.464	0.483	0.499
	T_c =	1019	1061	1099	1134	1198	1255	1306	1353	1397
n = 1.35	p_c =	67.8	81.2	95.2	109.8	140.4	172.9	207.1	242.7	279.8
	E =	0.355	0.384	0.409	0.431	0.466	0.494	0.517	0.537	0.553
	T_c =	1085	1137	1185	1230	1311	1383	1449	1510	1567
n = 1.41	p_c =	73.1	88.3	104.2	120.9	156.4	194.3	234.6	276.9	321.3
	E =	0.402	0.434	0.460	0.483	0.520	0.550	0.574	0.594	0.611
	T_c =	1170	1236	1297	1354	1459	1554	1642	1723	1799

*If, in any actual case, the real suction pressure should be some other value than $p_s = 12.5$ lb. per sq. in., as p'_s, the values of p_c in the above table can be corrected in the ratio $\frac{p'_s}{12.5}$.

actually a pound or more below the atmospheric pressure, that difference being necessary to open the air-admission valve (when automatic) and to cause the air to flow in with sufficient velocity. The charge, moreover, is heated by contact with the cylinder walls and with the hot gases remaining in the clearance. The compression is usually polytropic, not adiabatic, because it occurs in a cast-iron cylinder, which takes heat from the gas while it is being compressed, and so makes the final temperature and pressure less than that calculated on the assumption of adiabatic expansion.

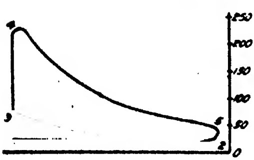


Fig. 8. Indicator Card from Otto-Cycle Engine

Table II shows the effect of the clearance on the efficiency for various values of the polytropic exponent (n) of the compression line.

Effect of Tardy Explosions. The explosion in the real engine is neither instantaneous nor complete. It approximates more closely to the ideal explosion when the compression is considerable, and when the explosive mixture has only a small excess of air present. Gasoline engines show a more rapid explosion than gas engines, and have indicator cards, as in Fig 9. The clearances are generally larger, so as to avoid excessive pressures. With weaker mixtures, the explosion becomes slower and less complete, as shown in Fig. 10, until, with the weakest explosive mixture, the process is really one of slow combustion taking place throughout the whole of the expansion period, and some of the charge may be still unburned when exhaust takes place. The fact that the clearance volume is full of products of combustion from the previous explosion, when the charge is drawn in, has a delaying effect upon the explosion, since the explosive mixture is diluted by the inert gases left in the clearance. Even under the best conditions, the rise of temperature, and consequently of pressure, during the explosion, is only about .6 of that occurring in the ideal engine. This, it will be seen, makes the power of the real engine considerably less than that of the ideal. The water jacket around the cylinder, without which the cylinder would be too hot to be properly lubricated, is one of the important causes of the difference between the real and ideal cycles, as the jacket usually absorbs about 40 per cent of the total heat of the combustion.

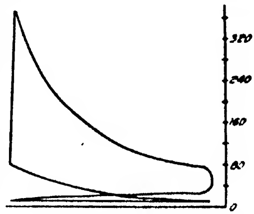


Fig 9 Indicator Card of Gasoline Engine

Fig 10. Indicator Card with Weak Mixture

Influence of Change of Specific Heats of Gases. In the ideal cycle the assumption is made that the specific heats of the working substance remain constant throughout the temperature range existing in the cycle. This is erroneous,

TABLE III

Effect of Variable Specific Heats on the Theoretical Thermal Efficiency

Volumetric Compression Ratio	Theoretical Thermal Efficiency	
	Specific Heats, Constant	Specific Heats, Variable
3	35.5	28.25
4	42.75	34.90
5	47.75	39.50
6	51.5	43.0
7	54.25	45.5

as the specific heats vary materially with change of temperature. This is one of the most important causes why the pressure and temperature at the end of explosion fall below those calculated for the ideal cycle. The effect of this variability of specific heats on the theoretical thermal efficiency is shown in Table III, which is based on an average of the results of Clerk and Wimperis.

The expansion curve is above the adiabatic in real engines, because the cylinder walls, which have been heated by the explosion, give back some heat to the gases, and also because the combustion still continues and liberates more heat. This last effect is especially marked when the explosive mixture is weak.

Early Beginning of Exhaust. Finally, the exhaust, as in the steam engine, begins a little before the end of the expansion stroke, so as to give plenty of time for the escape of the gases. Consequently, the card, at the point *5*, Fig. 8, is very much rounded off, and the pressure in the cylinder during the exhaust stroke is necessarily higher than that of the atmosphere into which the gases are rejected.

Final Efficiency. The total effect of all these differences between the real and the ideal engine is that the work done in an actual engine in good condition is only from .3 to .75 of that which the ideal engine would do; or, in other words, the efficiency of the real engine is only from .3 to .75 that of the ideal engine, depending upon the fuel used.

Use of Reducing Factor in Real Calculations. **The actual values of the explosion pressure and temperature can be found by multiplying the absolute pressure and temperature taken from the ideal, or**

hypothetical, card, by a reduction factor which takes into account the decrease in temperature or pressure due to heat losses, cooling, etc. The value of this factor is approximately that of the card factor, or ratio between the efficiencies of the real and ideal engine.

Example. What are the probable actual explosion pressure and temperature in the engine of the preceding examples if the real engine has .6 the efficiency of the ideal engine?

Solution. The explosion pressure of the ideal engine was found to be

$$P_4 = 628.6 \text{ lb. per sq. in., absolute}$$

Therefore

$$\text{Probable actual } P_4 = 628.6 \times 0.6 = 377.2 \text{ lb. per sq. in., absolute}$$

The explosion temperature of the ideal engine was found to be

$$T_4 = 5568.6° \text{ absolute}$$

Therefore

$$\text{Probable actual } T_4 = 5568.6 \times 0.6 = 3341.2° \text{ absolute}$$

or

$$\text{Probable actual } t_4 = 2880.2° \text{ F.}$$

Example. What are the probable actual efficiency, horsepower, and gas consumption of the engine whose ideal performance has been worked out in the preceding examples? Assume the real engine to have .6 the efficiency of the ideal engine.

Solution.

The ideal efficiency was found to be .433

Therefore

$$\text{Probable real efficiency} = .6 \times .433 = .26$$

The ideal horsepower was found to be 120.3

Therefore

$$\text{Probable real h.p.} = .6 \times 120.3 = 72.2, \text{ nearly}$$

The gas consumption is expressed in cubic feet per i.h.p. per hour. In the ideal engine the volume of gas taken in per cycle was

$$\frac{V_2 - V_1}{L+1} = \frac{1.178}{8} = .147 \text{ cu. ft.}$$

The number of cycles per minute was 125

Therefore

$$\text{Gas used per minute} = .147 \times 125 = 18.4 \text{ cu. ft.}$$

$$\text{Gas used per hour} = 18.4 \times 60 = 1104 \text{ cu. ft.}$$

And the probable real i.h.p. is 72.2

Therefore

$$\text{Gas used per i.h.p. per hour} = \frac{1104}{72.2} = 15.29 \text{ cu. ft.}$$

EXAMPLES FOR PRACTICE

What are the probable actual efficiency, i.h.p., and gas consumption of the engine whose ideal performance has been worked

out in the previous examples for practice; efficiency of real engine .6 of efficiency of ideal.

$$\text{Ans.} \begin{cases} .287 \text{ efficiency} \\ 108 \text{ i.h.p.} \\ 14.71 \text{ cu. ft. gas consumption} \end{cases}$$

Indicated Horsepower. The indicated horsepower of the normal Otto-cycle gas engine is determined from the area of the indicator card, just as with the steam engine; but there are some special points to which attention must be paid. In the usual formula,

$$\text{i.h.p.} = \frac{PLAN}{33,000}$$

N is the number of cycles per minute, not the number of revolutions. The mean effective pressure P is obtained from the indicator card by going around it with a planimeter in the way in which it was traced—that is, in the order *1-2-3-4-5-1*, Fig. 8. The indicator card consists really of two areas or loops, of which *3-4-5* represents positive work, and *1-2* negative work. The total work done on the piston is represented by the difference between these two areas. The small area *1-2* represents the work done in overcoming the friction resistance of the gas when being admitted to and expelled from the cylinder. It is work which has to be done by the engine, is a definite loss of power, and should be made as small as possible. The area *3-4-5* is the work which is actually done on the piston, less the work required to compress the gas; it is the true work of the cycle, all of which would be available for driving the engine were it not for the gas-friction resistances represented by the area *1-2*. If a planimeter is made to trace the diagram in the order in which it was drawn, it will go around the areas *1-2* and *3-4-5* in opposite directions; that is, if it goes around the one clockwise, it will go around the other contra-clockwise. The consequence will be that the readings of the planimeter will give the desired difference between the two areas *3-4-5* and *1-2*. The mean effective pressure is then obtained from this area in the usual manner.

DIESEL CYCLE

Characteristics of the Cycle. The first cycle proposed by Diesel consisted of isothermal or constant temperature compression, fol-

lowed by a further adiabatic compression, isothermal combustion, and adiabatic expansion to atmospheric pressure. Practical difficulties, however, led to modifications of this proposed cycle until the actual cycle of today has little in common with it. The first modification of the proposed cycle was to carry on the compression adiabatically, as in any gas engine. It was attempted to carry on the combustion isothermally, but in order to approximate this the combustion must be externally regulated, so that just enough heat will be generated at each instant to keep the temperature constant as long as fuel is being injected. In the Diesel engine, as at present constructed, no such control is attempted. Indicator cards from recent Diesel engines show that the combustion takes place nearly at constant pressure.

Ideal Cycle. The processes taking place in the cylinder are represented on the pressure-volume diagram, or ideal indicator card, Fig. 11. Exhaust, suction, and compression in this cycle are precisely the same as in the Otto cycle and are represented on the pressure-volume diagram in the same way. Up to this point, the only differences between the Diesel and the Otto cycles are: first, in the Otto cycle it is *assumed* for convenience that pure air alone is admitted and compressed, and in the Diesel cycle this is actually the case; second, the compression in the Diesel cycle is carried much higher than in the Otto.

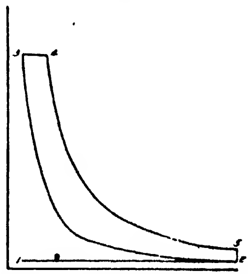


Fig. 11 Ideal Indicator Card of Diesel Cycle

When the charge of pure air has been compressed until it occupies the clearance volume, represented by point *3*, oil is injected, in the form of a fine spray, into the hot compressed air which vaporizes and completely burns the oil as it enters. At this point the piston starts on its outstroke, and, theoretically, oil is injected and burned at just the rate to counteract the fall in pressure which would naturally accompany this increase in volume and maintain the pressure constant along the line *3–4* (heat is added at constant pressure). At the point *4* the fuel injection is cut off and the hot products of com-

bustion at the pressure P_4 force the piston out, and, expanding adiabatically behind it, fall in pressure and temperature. As in the Otto cycle, the expansion curve *4–5* is similar to the compression curve *2–3*, and has the same equation.

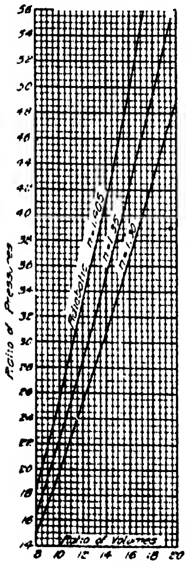


Fig. 12. Polytropic Expansion or Compression Curves of Fig. 6 Extended to Include Ratios Used in Diesel Cycle

From this point on, the events are the same as in the Otto cycle. In the Otto cycle, heat is received and rejected at constant volume; while in the Diesel cycle, heat is received at constant pressure and rejected at constant volume.

Pressures and Temperatures During the Cycle. *Pressure and Temperature after Compression.* The pressure and temperature at the point *3* can be found, as in the case of the Otto cycle from the curves, Fig. 12. The curves of Fig. 12 are exactly similar to those of Fig. 6, in fact, they are the curves of Fig. 6 extended to include the ratio of volumes obtaining in the Diesel cycle, while the ratios of Fig. 6 cover the case of the Otto cycle.

Since Diesel motors, as constructed, are oil engines, the increase of volume due to the injection of fuel, i. e., the increase of charge weight, may be neglected without serious error since the volume occupied by the fuel vapor in explosive mixtures of air and liquid-fuel vapors is negligible. (For benzine vapor the ratio is 45 to 1, and even in the case of crude alcohol the volume of the alcohol vapor occupies only between 2 and 3 per cent of the volume of the mixture.) If a gas, especially a lean gas, were under consideration, this assumption would not be permissible. The volume of mixture of air and fuel vapor per pound of fuel may therefore be taken, for maximum load conditions, as the volume of air (L) actually necessary per pound of fuel to give best combustion results.

Rise in Temperature During Combustion. The rise in temperature during combustion depends on how much heat is generated, which, in turn, depends on the amount of fuel injected and the heat of combustion of a pound of the fuel. The amount of the fuel injected for maximum load depends on the total volume of pure air taken into the cylinder at each suction stroke, since no more fuel should be injected than can be burned with the best results; therefore

$$\text{Weight of fuel per cycle} = \frac{V_2 - V_1}{L}, \text{ pounds}$$

and the heat of combustion of this fuel is

$$Q = \frac{H(V_2 - V_1)}{L} \text{ B.t.u.}$$

where H is the heat of combustion of a pound of the fuel in B.t.u. This heat is utilized in raising the temperature of the air from the known temperature T_3 to an unknown temperature T_4. The rise in temperature can be found if the heat necessary to raise one pound of air one degree in temperature while the pressure remains constant is known. This amount of heat is called the *specific heat at constant pressure*, and is denoted by the symbol C_p. C_p for air equals 0.237 B.t.u. With a weight of w pounds of air per charge, the heat necessary to raise the temperature of the charge from T_3 to T_4 is

$$wC_p(T_4 - T_3) \text{ B.t.u.}$$

and the heat of combustion is used entirely in raising the temperature of the charge from T_3 to T_4; hence

$$\frac{H(V_2 - V_1)}{L} = wC_p(T_4 - T_3)$$

$$= \left(\frac{P_2V_2}{RT_2}\right)C_p(T_4 - T_3)$$

$$T_4 - T_3 = \left(\frac{H}{L}\right)\left(\frac{RT_2}{P_2}\right)\left(\frac{1}{C_p}\right)\frac{(V_2 - V_1)}{V_2}$$

Example. A Diesel engine with 7.14 per cent clearance, taking in its charge of air at atmospheric pressure and temperature (14.7 pounds per square inch and 60° F.), is supplied with crude oil with a heat value of 18,000 B.t.u. per pound and requiring 300 cubic feet of atmospheric air per pound of fuel to give best combustion at maximum load. What are (1) the pressure and (2) the temperature at the end of compression, and (3) the temperature at the end of combustion when

(*a*) the compression is adiabatic, $n = 1.405$

(*b*) the compression is polytropic, $n = 1.35$

Solution. (1) The clearance volume V_3 is 7.14 per cent of the volume $V_2 - V_3$, through which the piston moves; or

$$V_3 = (V_2 - V_3)\frac{7.14}{100}$$

hence

$$100V_3 = 7.14V_2 - 7.14V_3$$

and

$$107.14V_3 = 7.14V_2$$

$$\frac{V_2}{V_3} = \frac{107.14}{7.14} = 15.0$$

From the curves, Fig. 12, if the ratio of compression is 15:

(*a*) The corresponding ratio of pressures is 45.53, and the pressure at the end of the compression

$$P_3 = 45.53 \times 14.7 = 669.3 \text{ lb. per sq. in., absolute}$$

(*b*) The ratio of pressures is 38.70, and

$$P_3 = 38.70 \times 14.7 = 568.9 \text{ lb. per sq. in., absolute}$$

(2) $$T_3 = P_3\left(\frac{V_3}{V_2}\right)\left(\frac{T_2}{P_2}\right)$$

(*a*) $$T_3 = 669.3 \times 144\left(\frac{1}{15}\right)\left(\frac{521}{14.7 \times 144}\right) = 1581° \text{ absolute}$$

$$t_3 = 1120° \text{ F.}$$

(*b*) $$T_3 = \frac{568.9 \times 144 \times 1 \times 521}{(15 \times 14.7 \times 144)} = 1344° \text{ absolute}$$

$$t_3 = 883° \text{ F.}$$

(3) $$T_4 - T_3 = \left(\frac{H}{L}\right)\left(\frac{RT_2}{P_2}\right)\left(\frac{1}{C_p}\right)\frac{(V_2 - V_3)}{V_2}$$

$$\frac{(V_2 - V_3)}{V_2} = \frac{\left(V_2 - \frac{V_2}{15}\right)}{V_2} = \frac{14}{15}$$

$$T_4 - T_3 = \left(\frac{18,000}{300}\right)\left(\frac{53.2 \times 521}{14.7 \times 144}\right)\left(\frac{1}{0.237}\right)\left(\frac{14}{15}\right)$$

hence

$$T_4 = 3094 + T_3° \text{ absolute}$$

$$t_4 = 3094 + t_3° \text{ F.}$$

(*a*) $t_4 = 3094 + 1120 = 4214° \text{ F.}$

(*b*) $\quad = 3094 + 883 = 3977° \text{ F.}$

Volumes at End of Combustion. If a perfect gas is raised in temperature while its pressure is unchanged, the volume will increase in exact proportion to the rise of absolute temperature; or

$$V_4 : V_3 :: T_4 : T_3$$

hence

$$V_4 = \left(\frac{T_4}{T_3}\right)V_3$$

Example. What are the volumes at the end of combustion in the preceding problem, expressed as ratios of the clearance volume?

Solution.

$$(a)\quad \frac{V_4}{V_3}=\left(\frac{4214+461}{1581}\right)=2.957$$

$$(b)\quad =\left(\frac{3077+461}{1344}\right)=3.302$$

These results are checked by simple reasoning, since it is self-evident that in order to add the same amount of heat at constant pressure, the lower the initial temperature the greater must be the volume change.

In the Otto cycle the ratio of compression $\frac{V_2}{V_3}$ and the ratio of expansion $\frac{V_5}{V_4}$ are the same, but in the Diesel cycle this is not the case. In the Diesel cycle V_5 equals V_2 and V_4 equals dV_3, where d is the ratio between the cut-off volume V_4 and the clearance volume V_3, or $\frac{V_4}{V_3}$.

Influence of Early Cut-Off. The case just described, in which as much fuel is injected as is possible in order to get the best possible combustion results with the volume of air present, corresponds to the maximum possible load of the engine. Since an engine must be capable of carrying an overload, the normal rated load must be carried when less than the maximum possible amount of fuel is injected. In actual practice the cut-off occurs at about 10 per cent of the stroke at normal rated load and, with a compression ratio of 16, the cut-off ratio d is, at maximum possible load, about 3.0 and at normal load 2.5. To prove the latter value

$$\left(\frac{V_4-V_3}{V_2-V_1}\right)=\frac{1}{10}\quad\text{and}\quad\frac{V_2}{V_1}=16$$

$$(V_4-V_3)=\frac{(V_2-V_1)}{10}$$

$$\left(\frac{V_4}{V_3}\right)-1=\frac{(V_2-V_1)}{10V_3}=\frac{V_2}{10V_3}-\frac{V_1}{10V_3}$$

but

$$V_3=V_1$$

hence

$$\frac{V_4}{V_3}-1=\frac{V_2}{10V_1}-\frac{1}{10}$$

$$d=\frac{V_4}{V_3}=\frac{16}{10}-\frac{1}{10}+1=2.5$$

A reduction to any extent in the fuel injected below the maximum is possible because of the fact that the oil is burned as it enters, like gas in a burner, so that it can be properly accomplished with any excess of air.

When the amount of fuel injected is less than the maximum possible, the temperature at the end of combustion is

$$T_4 = d\,T_3$$

and the amount of heat added per cu. ft. of piston displacement is

$$Q = (T_4 - T_3)\left(\frac{P_2}{RT_2}\times C_p\frac{V_2}{(V_2 - V_1)}\right)\text{ B.t.u.}$$

Example. If the engine of the previous problem cuts off at 10 per cent of the stroke at normal load, what (1) is the temperature at the end of combustion, and (2) the heat added per cubic foot of piston displacement?

Solution.

$$d = \frac{V_4}{V_3} = \frac{1}{10}\left(\frac{V_2}{V_1}\right) + \frac{9}{10}$$

$$= \frac{15}{10} + \frac{9}{10} = 2.4$$

$$T_4 = dT_3$$

(1) (a) $T_4 = 2.4 \times 1581 = 3794°$ absolute. $t_4 = 3333°$ F.

(b) $= 2.4 \times 1344 = 3226°$ absolute. $= 2765°$ F.

(2) $Q = (T_4 - T_3)\left(\frac{14.7\times 144 \times 0.237 \times 15}{53.2 \times 521 \times 14}\right)$

$= (T_4 - T_3) \times 0.01939$

(a) $= (3794 - 1581) \times 0.01939 = 42.92$ B.t.u. per cu. ft. of piston displacement

(b) $= (3226 - 1344) \times 0.01939 = 36.51$ B.t.u. per cu. ft. of piston displacement

Ratio of Expansion. The ratio of expansion in the Diesel cycle is the ratio of compression divided by the ratio of cut-off volume to clearance volume, since, as the ratio of expansion equals $\frac{V_5}{V_4}$, and V_5 equals V_2, and V_4 equals dV_3
hence

$$\frac{V_5}{V_4} = \left(\frac{V_2}{V_3}\right)\left(\frac{1}{d}\right) = \frac{\text{ratio of compression}}{d}$$

If this ratio of expansion is known, the pressure and temperature at the end of expansion can be found in exactly the same manner as in the case of the compression.

Example. In the engine of the previous problems, what is (1) the pressure and (2) the temperature at the end of expansion if (I) the engine is developing the maximum possible power and (II) is operating at normal load?

Solution.

$$\frac{V_5}{V_4}=\left(\frac{V_2}{V_3}\right)\left(\frac{1}{d}\right)$$

(I) (a) (1) $$=\frac{15}{2.957}=5.08$$

$$P_4=P_3=669.3 \text{ lb. per sq. in., absolute (see page 30)}$$

Referring to the curves of Fig. 6, if the working substance expands adiabatically to 5.08 times its original volume, the pressure falls to $\frac{1}{9.7}$ of its original value. Therefore

$$P_5=\frac{669.3}{9.7}=69.0 \text{ lb. per sq. in., absolute}$$

(2) $$\frac{(P_4V_4)}{T_4}=\frac{(P_5V_5)}{T_5}$$

$$T_5=\left(\frac{P_5}{P_4}\right)\left(\frac{V_5}{V_4}\right)T_4$$

$$=\left(\frac{1}{9.7}\right)\times 5.08\times(4214+461)$$

$$=2449° \text{ absolute} \qquad t_5=1988° \text{ F.}$$

(b) (1) $$\frac{V_5}{V_4}=\frac{15}{3.302}=4.54$$

$$P_5=\frac{568.9}{7.75}=73.4 \text{ lb. per sq. in., absolute}$$

(2) $$T_5=\frac{1}{7.75}\times 4.54\times(3977+461)$$

$$=2600° \text{ absolute} \qquad t_5=2139° \text{ F.}$$

(II) (a) (1) $$d=2.4$$

$$\frac{V_5}{V_4}=\frac{15}{2.4}=6.25$$

$$P_5=\frac{669.3}{13.05}=51.3 \text{ lb. per sq. in., absolute}$$

(2) $$T_5=\frac{1}{13.05}\times 6.25\times 3794$$

$$=1818° \text{ absolute} \qquad t_5=1357° \text{ F.}$$

(b) (1) $$P_5=\frac{568.9}{11.85}=48.0 \text{ lb. per sq. in., absolute}$$

(2) $$T_5=\frac{1}{11.85}\times 6.25\times 3226$$

$$=1702° \text{ absolute} \qquad t_5=1241° \text{ F.}$$

Efficiency. In the Diesel cycle, the heat added has been seen to be

$$Q=wC_p(T_4-T_3) \text{ B.t.u.}$$

Heat is rejected from the engine only during the process represented by the line *5–2*, because when the charge gets back to the

condition 2, it has returned to its original volume and pressure, and consequently to its original temperature. The heat rejected is then

$$Q_r = wC_v\,(T_5 - T_2)\ \text{B.t.u.}$$

and the work done per cycle is

$$W = Q - Q_r\ \text{B.t.u.} = 778\ (Q - Q_r)\ \text{ft.-lb.}$$

The *efficiency of the cycle*—that is, the fraction of the heat supplied that is converted into work—is

$$E = \frac{W}{Q} = \frac{Q - Q_r}{Q} = 1 - \frac{Q_r}{Q}$$

$$= 1 - \frac{wC_v(T_5 - T_2)}{wC_p(T_4 - T_3)}$$

$\frac{C_v}{C_p}$ which is the reciprocal of the ratio of the specific heats (C_v is the specific heat at constant volume $= 0.169$ for air), equals the reciprocal of the exponent for adiabatic expansion or compression, or $\frac{1}{1.405}$, and since, as already stated

$$d = \frac{T_4}{T_3}$$

and

$$\frac{T_5}{T_2} = \frac{P_5}{P_2}\ \text{(constant volume)}$$

$$\frac{P_5}{P_2} = \frac{\left(\dfrac{P_4V_4^n}{V_5^n}\right)}{\left(\dfrac{P_3V_3^n}{V_2^n}\right)} = \left(\frac{V_4}{V_3}\right)^n = d^n$$

since

$$P_4 = P_3 \qquad \text{and } V_2 = V_5$$

hence

$$T_5 = T_2 d^n$$

therefore

$$\frac{(T_5 - T_2)}{(T_4 - T_3)} = \frac{(T_2 d^n - T_2)}{(dT_3 - T_3)}$$

$$= \frac{T_2(d^n - 1)}{T_3(d - 1)}$$

therefore

$$E = 1 - \left(\frac{1}{1.405}\right)\left(\frac{T_2}{T_3}\right)\left(\frac{d^n - 1}{d - 1}\right)$$

From this it is seen that the efficiency of the Diesel cycle depends, not only on the compression, but also upon d, i. e., the volume at the end of combustion. The smaller the value of d (the earlier in the stroke the fuel supply is cut off) the greater is the thermal efficiency, other conditions remaining the same. This fact is borne out in practice within limits, as a large number of tests of Diesel engines have shown a greater thermal efficiency at three-quarters than at full load. That this does not hold for still lower loads is due to the influence of other factors.

This equation for efficiency holds only when both the compression and expansion lines have the same exponent. If the exponents are different, the efficiency must be found from the quantities of heat added and rejected.

Example. If the engine of the preceding problems has an exponent for the compression line of $n=1.35$, and for the expansion line of $n=1.41$, what is the efficiency of the cycle at maximum load?

Solution.

$$E=1-\frac{Q_r}{Q}$$

$$Q=H\frac{(V_1-V_2)}{L}$$

$$=18{,}000\,\frac{(V_1-V_2)}{300}\ \text{B.t.u. per cycle}$$

$$Q_r=wC_v(T_4-T_1)=\left(\frac{P_1V_1}{RT_1}\right)C_v(T_4-T_1)$$

$$\frac{Q_r}{Q}=\frac{P_1C_v(T_4-T_1)}{60RT_1}\times\frac{V_1}{V_1-V_2}$$

From the results of the preceding problems, when the compression line had the exponent $n=1.35$,

$$T_3=4438°\ \text{absolute.}\qquad \frac{V_1}{V_1-V_2}=\frac{15}{14}\qquad d=3.302\quad \text{(See p. 31)}$$

$$\frac{V_4}{V_3}=\left(\frac{V_1}{V_2}\right)\left(\frac{1}{d}\right)=\frac{15}{3.302}=4.54$$

Since the expansion is adiabatic, the pressure ratio must be found from the adiabatic curve of Fig. 6, and is, for a volume ratio of $\frac{V_4}{V_3}=4.54$, equal to $\frac{1}{8.3}$, therefore

$$T_4=\left(\frac{P_4}{P_3}\right)\left(\frac{V_4}{V_3}\right)T_3$$

$$=\left(\frac{1}{8.3}\right)(4.54)\times 4438=2428°\ \text{absolute}$$

$$\frac{Q_r}{Q}=\frac{14.7\times 144\times 0.169\,(2428-521)}{60\times 53.2\times 521}\times\frac{15}{14}=0.440$$

$$E=1-0.440=0.560$$

Use of Efficiency Factor Curves. From Fig. 13, the value of th factor $\frac{d^n-1}{1.405\,(d-1)}$ can be obtained for the three usual values of n an any value of d which is to be met with in practice. With the ai of these curves the efficiency for any given case can be determined

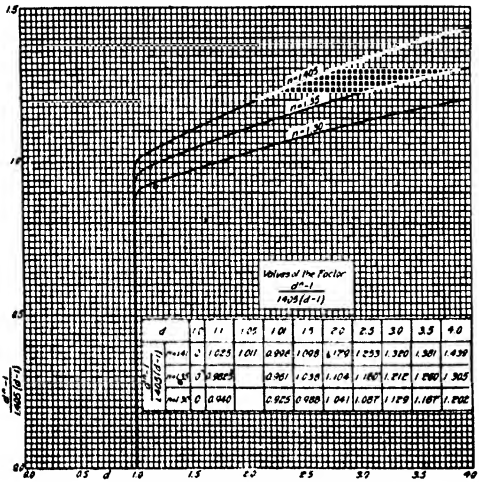


Fig. 13. Efficiency Factor (F) for Diesel Cycle

Example. Find the efficiencies of the cycles in the preceding problem.

Solution.

$$E=1-\frac{T_1}{T_1}\times\frac{d^n-1}{1.405\,(d-1)}=1-\frac{T_1}{T_1}\times F$$

$$T_1=60+461=521^\circ \text{ absolute}$$

(a) $T_1=1581^\circ$ absolute

(b) $=1344^\circ$ absolute

(I) Under maximum load conditions

(a)	$d=2.957$	$n=1.405$	$F=1.319$
(b)	$d=3.302$	$n=1.35$	$F=1.243$

(II) Under normal load conditions $d=2.4$

(a) $n=1.405$ $F=1.241$

(b) $n=1.35$ $F=1.154$

Solution.

(I) (a) $E=1-\frac{521}{1581}\times 1.319$

$=1-0.434=0.566$

(b) $E=1-\frac{521}{1344}\times 1.243$

$=1-0.482=0.518$

(II) (a) $E=1-\frac{521}{1581}\times 1.241$

$=1-0.409=0.591$

(b) $E=1-\frac{521}{1344}\times 1.154$

$=1-0.447=0.553$

Work Done per Cycle. The work done per cycle can be calculated from the efficiency without knowing the heat rejected.

$$E=\frac{W}{Q}$$

or

$$W=E\times Q \text{ B.t.u.}=778\times E\times Q \text{ ft.-lb.}$$

Example. If the engine of the preceding problems has a cylinder of 12 inches diameter and 18 inches stroke, what will be the work done per cycle?

(I) Under maximum load conditions

(II) Under normal load conditions

Solution.

$$V_2-V_1=\frac{\pi}{4}\times\left(\frac{12}{12}\right)^2\times\frac{18}{12}=1.178 \text{ cu. ft.}$$

(I) $Q=H\left(\frac{V_2-V_1}{L}\right)$

$=\frac{18,000}{300}\times 1.178=70.68$ B.t.u.

$=778\times 70.68=54,990$ ft.-lb.

(a) $W=54,990\times 0.567=31,180$ ft.-lb.

(b) $=54,990\times 0.519=28,540$ ft.-lb.

(II) (a) $Q=42.92$ B.t.u.

$W=778\times 42.92\times 0.592=19,770$ ft.-lb.

(b) $Q=36.51$ B.t.u.

$W=778\times 36.51\times 0.554=15,740$ ft.-lb.

Since this engine requires two revolutions to a complete cycle, the number of cycles per minute is only half the number of revolutions per minute.

Example. If the engine makes 250 r.p.m., what will be the i.h.p.?

Solution. Work per minute $= W \times 125$

$$\text{Horsepower} = \frac{W \times 125}{33,000} = \frac{W}{264}$$

(I) (a) $= \frac{31,180}{264} = 118.1$ i.h.p.

(b) $= \frac{28,540}{264} = 108.1$ i.h.p.

(II) (a) $= \frac{19,770}{264} = 74.9$ i.h.p.

(b) $= \frac{15,740}{264} = 59.6$ i.h.p.

EXAMPLES FOR PRACTICE

1. A Diesel engine has 7.69 per cent clearance and takes in its charge of air at 14.7 lb. per square inch and at 60° F. What is (1) the pressure, and (2) the temperature at the end of the compression when

(a) the compression is adiabatic, and $n = 1.405$

(b) the compression is polytropic, and $n = 1.30$

Ans. (a) (1) 607.1 lb. per sq. in., absolute
(2) 1076° F.
(b) (1) 454.1 lb. per sq. in., absolute
(2) 689° F.

2. If this engine is supplied with crude oil, with a heat value of 18,600 B.t.u. per pound and requiring 300 cubic feet of atmospheric air per pound of fuel to give best combustion results, what is the temperature under maximum load conditions at the end of combustion? Ans. (a) 4157° F.; (b) 3770° F.

3. What are (1) the pressure and (2) the temperature at the end of expansion under the conditions of the previous problem?

Ans. (a) (1) 59.4 lb. per sq. in., absolute
(2) 1999° F.
(b) (1) 78.3 lb. per sq. in., absolute
(2) 2317° F.

4. If the engine cuts off at 10 per cent of the stroke at normal load, what (1) is the temperature at the end of combustion, and (2) the heat added per cubic foot of piston displacement?

Ans. (1) (a) 3074° F.
(b) 2184° F.
(2) (a) 38.94 B.t.u. per cu. ft.
(b) 29.14 B.t.u. per cu. ft.

5. Under the conditions of problem 4, what are (1) the pressure and (2) the temperature at the end of expansion?

Ans. $\begin{cases} (1) & (a) & \text{48.4 lb. per sq. in., absolute} \\ & (b) & \text{43.0 lb. per sq. in., absolute} \\ (2) & (a) & 1254^\circ \text{ F.} \\ & (b) & 1065^\circ \text{ F.} \end{cases}$

6. What is the efficiency of the cycle under (1) the maximum load conditions and (2) normal load conditions?

Ans. $\begin{cases} (1) & (a) & 0.551 \\ & (b) & 0.477 \\ (2) & (a) & 0.574 \\ & (b) & 0.520 \end{cases}$

7. If the cylinder diameter is 18 inches and the stroke 24 inches, and the engine makes 200 revolutions per minute, what is the i.h.p. (1) under maximum load conditions and (2) under normal load conditions?

Ans. $\begin{cases} (1) & (a) & \text{275.8 i.h.p.} \\ & (b) & \text{238.9 i.h.p.} \\ (2) & (a) & \text{186.4 i.h.p.} \\ & (b) & \text{126.4 i.h.p.} \end{cases}$

Ideal and Real Diesel Cycles. *Factors Showing Differences.* Fundamentally, the modification of the theoretical Diesel cycle does not differ in practice from that already stated for the Otto cycle, as far as the suction, compression, expansion, and exhaust lines are concerned. The combustion line, since the combustion takes place at constant pressure rather than at constant volume, needs further attention.

The injection of fuel is never exactly so regulated as to develop a combustion line showing exactly constant pressure, the line, in practice, showing a diminution of pressure as the piston moves forward and the combustion proceeds. On the other hand, the combustion is never found to take place isothermally, as was proposed by Diesel in the first change which he made in his cycle. Güldner found from analysis of indicator diagrams that there was a decided temperature increase along the combustion line, in spite of the fact that it looked isothermal. He found that the air was compressed to approximately 1100° F. and that at the full cut-off this had increased to roughly 2700° F. Due to afterburning, the maximum temperature was about 270° F. higher than this and occurred after the cut-off, so that it can

be seen from this that afterburning affects the combustion and expansion lines in the Diesel cycle, as well as in the Otto cycle, but to a less extent.

These results should be compared with the calculated results on pages 30 and 32 for the case where the exponent of the compression line was 1.35—the average exponent of compression lines found in practice—and the engine cut-off at 10 per cent of the stroke.

Effects of Departures from Ideal Conditions. The total effect of these differences between the real and the ideal engine is that the work done in an actual Diesel engine in good condition is only from 50 per cent to 75 per cent of that which the ideal engine (compression and expansion adiabatic) would do, or, in other words, the efficiency of the real engine is only from 50 per cent to 75 per cent of that of the ideal engine, depending on the design and workmanship; properly designed and constructed engines average about 70 per cent.

Example. What are the probable actual efficiency, horsepower, and oil consumption of the ideal engine of the preceding illustrative examples (1) under maximum load conditions, and (2) under rated load conditions, assuming the real engine to have .7 the efficiency of the ideal cycle?

Solution. When the engine compresses and expands the charge adiabatically, the ideal efficiency was found to be

(1) 0.566
(2) 0.591

Probable real efficiency

(1) $0.7\times.566=0.396$
(2) $0.7\times.591=0.414$

The ideal horsepower was found to be

(1) 118.1
(2) 74.9

Probable real horsepower

(1) $0.7\times118.1=82.7$
(2) $0.7\times74.9=52.4$

The oil consumption is expressed in pounds per i.h.p. per hour. In the ideal engine the volume of charge per cycle is V_2-V_1, and the volume of charge per pound of fuel at maximum load is L. Therefore, the weight of fuel injected per cycle at maximum load is

$$\frac{V_2-V_1}{L}=\frac{1.178}{300}=.00393 \text{ lb.}$$

At maximum load the heat added per cycle is

$$\frac{H}{L}=\frac{18,000}{300}=60 \text{ B.t.u.}$$

But at normal load the fuel admission is cut off, so that only 42.92 B.t.u. are

added. Therefore, the amount of fuel injected at normal load is only $\frac{42.92}{60}$ or 0.715 of that injected at maximum load. Therefore, the oil used per hour is

At maximum load, $0.00393\times125\times60=29.46$ lb.

At normal load, $29.46\times0.715=21.06$ lb.

The probable real i.h.p. is 82.7 and 52.4, respectively

Therefore, the oil consumption is

$$(1)\quad \frac{29.46}{82.7}=0.356 \text{ lb. per i.h.p. per hour}$$

$$(2)\quad \frac{21.06}{52.4}=0.402 \text{ lb. per i.h.p. per hour}$$

EXAMPLES FOR PRACTICE

What are the probable actual efficiency, i.h.p., and oil consumption of the engine whose ideal performance has been worked out in the previous examples for practice (1) at maximum load and (2) at normal load? Efficiency of the real engine is .7 of the efficiency of the ideal.

Ans. (1) 0.386 efficiency
193.2 i.h.p.
0.361 lb. of oil per i.h.p. per hour
(2) 0.402 efficiency
130.5 i.h.p.
0.352 lb. of oil per i.h.p. per hour

FUELS AND FUEL MIXTURES

COMPOSITION AND HEAT VALUES OF ENGINE FUELS

Classification of Gases. The fuels used in gas engines are extremely variable in origin, in composition, and in heat value. They consist almost entirely of the chemical elements carbon, hydrogen, and oxygen, and their compounds, diluted with more or less nitrogen. The intelligent appreciation of the method of manufacture and of the advantages of gaseous fuels in general, and producer gas in particular, necessitates a clear understanding of certain fundamental facts.

Gases may be divided into three classes: elementary, compound, and mixtures. Elementary gases consist of one element only—as oxygen, for instance. Compound gases are composed of two or more elements in chemical combination, such as marsh gas, in which

carbon and hydrogen are combined. Mixtures are not definite compounds, but consist of two or more elementary or compound gases simply mixed together without any chemical affinity existing between any of the constituents. Air is such a mixture, consisting essentially of 21 parts of oxygen and 79 parts of nitrogen by volume; or 23 parts of oxygen and 77 parts of nitrogen by weight.

Characteristics of Common Gases. The presence of certain desirable or undesirable constituents will give the gas desirable or undesirable properties, and these properties will be proportional to the relative percentage present of the constituents in question Hence, it is desirable to know the properties of each constituent, so that its effect on the gas as a whole may be determined.

Hydrogen. Hydrogen is the lightest known substance, is colorless, odorless, non-poisonous, very combustible, non-luminous, and burns with a pale blue flame.

Marsh Gas. Marsh gas, also called methane, is odorless, colorless, has a high calorific power but slow rate of combustion, and burns with a slightly luminous flame.

Olefiant Gas. Olefiant gas, also called ethylene or ethene, has a high calorific power, is colorless and odorless, and burns with a very luminous flame. It is sometimes spoken of as an "illuminant".

Carbon Monoxide. Carbon monoxide, also called carbonic oxide, is a deadly poison, colorless, odorless, insoluble in water, and burns with a blue flame.

Carbon Dioxide. Carbon dioxide, also called carbonic anhydride or carbonic acid, is soluble in water, odorless, colorless, and non-combustible.

Oxygen. Oxygen is colorless, tasteless, odorless, and its presence in gas decreases the amount of oxygen that must be furnished for combustion.

Nitrogen. Nitrogen is odorless, colorless, non-combustible, and has no effect in combustible mixtures except to act as a diluent.

Water Vapor. Water vapor comes from undecomposed steam passing through the fuel. On account of its high specific heat it may cause a large heat loss.

Physical Properties of Gases. *Volume.* The volume of a gas varies with the temperature and pressure. In order to secure comparable results from different tests, it is necessary that some

definite standard be used. The standard condition is taken as 62 degrees F. and a pressure of 29.92 inches of mercury.

Density. The density of a gas is the ratio of the weight of a unit volume of the gas to the weight of a unit volume of another gas taken as a standard, and at the same standard condition. Hydrogen and air are the standards usually used.

Thermal Capacity. The thermal capacity of a substance is the heat required to raise the temperature of a unit weight of it one degree. The *specific heat* of a gas is the ratio between the thermal capacities of equal weights of the gas and of water.

Sensible Heat. The sensible heat of a gas is the heat it carries by virtue of its temperature. The sensible heat per degree rise of temperature is equal to the product of the weight of gas and its specific heat.

B.T.U. The British thermal unit (B.t.u.) is the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit.

Calorific Power. The calorific power (or the heat value) of a gas is the number of heat units evolved by the complete combustion of a unit volume or weight of the gas.

High and Low Heat Values. When the heat value of a fuel gas is determined in a calorimeter, the products of combustion are cooled to the original temperature of the air-gas mixture. Most of the water vapor formed by the combustion of the free and combined hydrogen in the gas is condensed by this cooling. The measured quantity of heat, therefore, includes the latent heat of the water vapor (the heat given up by the water vapor in condensing from steam at the temperature of the mixture to water at the temperature of the mixture) recovered during this condensation.

The low heat value is the heat value of the gas when none of the water vapor formed by combustion has been condensed. This is the value on which gas engine efficiencies are usually based, since a gas engine cannot utilize the latent heat of the water vapor and therefore should not be charged with it.

The high heat value is the heat value of the gas when all of the water vapor formed by combustion has been condensed—but with none of the water vapor contained in the original gas or air condensed. To be sure that this condition obtains, the exhaust gas

TABLE IV

Combustion Products

Fuel	Symbol	Combustion Product	Volume After Combustion, per Cu. Ft. of Gas Burned	
			H_2O Cu. Ft.	CO_2 Cu. Ft.
Carbon monoxide	CO	$2CO + O_2 = 2CO_2$		1
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1	
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	2	1
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	3	2
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	2	2
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	1	2
Benzene	C_6H_6	$2C_6H_6 + 15O_2 = 12CO_2 + 6H_2O$	3	6
Hexane	C_6H_{14}	$2C_6H_{14} + 19O_2 = 12CO_2 + 14H_2O$	7	6
Nonane	C_9H_{20}	$C_9H_{20} + 14O_2 = 9CO_2 + 10H_2O$	10	9
Methyl alcohol	CH_4O	$2CH_4O + 3O_2 = 2CO_2 + 4H_2O$	2	1
Ethyl alcohol	C_2H_6O	$C_2H_6O + 3O_2 = 2CO_2 + 3H_2O$	3	2

must have the same humidity (water vapor content) as the original gas and air; otherwise this divergence must be corrected for.

CALCULATION OF PHYSICAL PROPERTIES OF A GASEOUS FUEL

Chemical and Physical Data. Table IV gives the names, chemical symbols, and combustion reactions of the various constituent gases contained in the gaseous fuels ordinarily used in engines. Table V gives the physical properties of these constituent gases. From these tables, and knowing the analysis—the percentages by volume or by weight of the constituents—of a particular gas, the physical properties of the mixture can be calculated.

Volumetric and Weight Analyses. The *volumetric analysis* can be determined from the analysis by weight by dividing the per cent by weight of each constituent present by the specific weight of that constituent (weight of one cubic foot under standard conditions) as given in Table V, and dividing each quotient by the sum of the quotients. The *analysis by weight* is determined in the reverse manner by multiplying the per cent of volume of each constituent by the specific weight and dividing each product by the sum of the products. Molecular weights may be used in place of specific weights.

Example. A typical producer gas made from anthracite coal has the following analysis by volume: CO_2, 5.2; O_2, 0.4; CO, 22.9; H_2, 15.3; CH_4, 1.0; N_2, 55.2. What is the analysis by weight?

TABLE V

Properties of Gases

Gas	Chemical Symbol	Molecular Weight	Specific Weight at 62° F. and 29.92 of Hg.	Density Air = 1	Specific Heat at		Required				Volume Contraction per Cu Ft. of Gas Burned	Heat Value of the Gas per Pound		Heat Value of the Gas per Cu Ft. at 62° F.	
					Constant Pressure	Constant Volume	Weight of		Volume of			High	Low	High	Low
							O_2 per Lb. of Gas	Air per Lb. of Gas	O_2 per Cu. Ft. of Gas	Air per Cu. Ft. of Gas					
			Lb. per Cu. Ft.		C_p	C_v	Lb.	Lb.	Cu. Ft.	Cu. Ft.	Cu. Ft.	B.t.u. per Lb.	B.t.u. per Lb.	B.t.u. per Cu. Ft.	B.t.u. per Cu. Ft.
Oxygen	O_2	32	0.0841	1.1054	0.218	0.156									
Nitrogen	N_2	28	0.0738	0.9674	0.244	0.173									
Air			0.0761	1.0000	0.237	0.169									
Steam gas	H_2O	18	0.0473	0.6222	0.481	0.373									
Carbon dioxide	CO_2	44	0.1156	1.5291	0.203	0.156									
Carbon monoxide	CO	28	0.0734	0.9670	0.243	0.173	0.572	2.477	0.5	2.38	0.5	4,380	4,380	322	322
Hydrogen	H_2	2	0.0053	0.0696	3.409	2.422	8.0	34.78	0.5	2.38	0.5	62,100	52,920	329	281
Methane	CH_4	16	0.0421	0.5535	0.589	0.460	4.0	17.32	2.0	9.52	0.0	23,850	21,650	1,003	913
Ethane	C_2H_6	30	0.0790	1.0380			3.733	16.16	3.5	16.7	−0.5	22,230	20,500	1,755	1,619
Ethylene	C_2H_4	28	0.0737	0.9852	0.37	0.30	3.429	14.85	3.0	14.3	0.0	21,600	20,420	1,591	1,504
Acetylene	C_2H_2	26	0.0684	0.8992	0.35	0.27	3.077	13.32	2.5	11.9	0.5	21,600	21,020	1,477	1,437
Benzene	C_6H_6	78	0.2052	2.698	0.299	0.213	3.077	13.32	7.5	35.8	−0.5	17,920	17,240	3,678	3,538
Hexane	C_6H_{14}	86	0.2262	2.973			3.585	15.37	9.5	45.3	−2.5	20,530	19,100	4,643	4,319
Nonane	C_9H_{20}	128	0.3364	4.425			3.5	15.22	14.0	66.7	−4.0	20,380	19,000	6,855	6,391
Methyl alcohol	CH_4O	32	0.0852	1.120	0.458	0.365	1.5	6.522	1.5	7.1	−0.5	9,500	8,400	810	716
Ethyl alcohol	C_2H_6O	46	0.1209	1.617	0.453	0.409	2.087	9.074	3.0	14.3	−1.0	13,330	12,180	1,612	1,473
Carbon	C	12	Burned	to CO								4,450	4,450		
			Burned	to CO_2								14,500	14,500		

Solution.

	Per cent by Vol		Specific Weight					Per cent by Weight
CO_2	5.2	×	0.1156	=	0.6011	× 100 ÷ 6.5126	=	9.2
O_2	0.4	×	0.0841	=	0.0336	× 100 ÷ 6.5126	=	0.5
CO	22.9	×	0.0734	=	1.6809	× 100 ÷ 6.5126	=	25.9
H_2	15.3	×	0.0053	=	0.0811	× 100 ÷ 6.5126	=	1.2
CH_4	1.0	×	0.0421	=	0.0421	× 100 ÷ 6.5126	=	0.6
N_2	55.2	×	0.0738	=	4.0738	× 100 ÷ 6.5126	=	62.6
	100.0				6.5126			100.0

Specific Weight. The specific weight can be determined by multiplying the volumetric per cent of each constituent present by the specific weight of that constituent as given in Table V, and dividing the sum of the products so obtained by 100.

Example. What is the specific weight of the gas of the preceding problem?

Solution. From the solution of the previous problem the sum of the products of the specific weights of the constituents by the volumetric percentage is 6.5126. Therefore, the specific weight of the producer gas is

$$\frac{6.5126}{100}=0.0651 \text{ lb. per cu. ft.}$$

Density. The density is determined by dividing the specific weight of the gas by the specific weight of air.

Example. What is the density of the producer gas of the previous problems?

Solution.

$$\text{Density}=\frac{0.0651}{0.0761}=0.8553$$

Specific Heat. The specific heat is determined by multiplying the per cent by weight of each constituent present by the specific heat of that constituent, as given in Table V, and dividing the sum of the products thus obtained by 100.

Example. What is the specific heat at constant pressure of the producer gas of the previous problems?

Solution.

	Per cent by Weight		C_p		
CO_2 =	9.2	×	0.203	=	1.868
O_2 =	0.5	×	0.218	=	0.109
CO =	25.9	×	0.243	=	6.294
H_2 =	1.2	×	3.409	=	4.091
CH_4 =	0.6	×	0.589	=	0.353
N_2 =	62.6	×	0.244	=	15.274
	100.0				27.989

$$\text{Specific heat}=\frac{27.989}{100}=0.280$$

The specific heat at constant volume is found in precisely the same manner, only using the values for the specific heats at constant volume of the constituents in place of the specific heats at constant pressure.

Weight of Oxygen or Air. The weight of oxygen or air chemically necessary for complete combustion is determined by multiplying the per cent by weight of each constituent present by the weight of oxygen or air required by that constituent, as given in Table V, and dividing the sum of the products so obtained by 100.

Example. What is the weight of air chemically necessary for the combustion of the producer gas of the previous problems?

Solution.

	Per cent by Weight		Weight of Air		
CO_2 =	9.2	×	0	=	0.00
O_2 =	0.5	×	−1	=	−0.5
CO =	25.9	×	2.477	=	64.14
H_2 =	1.2	×	34.78	=	41.74
CH_4 =	0.6	×	17.32	=	10.39
N_2 =	62.6	×	0	=	0.00
	100.0				115.77

$$\text{Weight of air required} = \frac{115.77}{100} = 1.16 \text{ lb. per lb. of gas}$$

Volume of Oxygen or Air. The volume of oxygen or air chemically necessary for complete combustion is determined by multiplying the per cent by volume of each constituent present by the volume of oxygen or air, as the case may be, required by that constituent, as given in Table V, and dividing the sum of the products thus obtained by 100.

Example. What is the volume of air chemically necessary for the combustion of the producer gas of the previous problems?

Solution.

	Per cent by Volume		Volume of Air		
CO_2 =	5.2	×	0.00	=	0.00
O_2 =	0.4	×	−1	=	−0.40
CO =	22.9	×	2.38	=	54.50
H_2 =	15.3	×	2.38	=	36.42
CH_4 =	1.0	×	9.52	=	9.52
N_2 =	55.2	×	0.00	=	0.00
	100.00				100.04

$$\text{Volume of air required} = \frac{100.04}{100} = 1.00 \text{ cu. ft. per cu. ft. of gas}$$

Excess of Air for Perfect Combustion. In practice, more air is used for combustion than the amount chemically necessary. If only the amount chemically necessary were used, the combustion would not be complete. The amount of excess air, therefore, is such as to give the most complete, or best, combustion and varies between 35 and 55 per cent over that chemically necessary.

Example. The producer gas of the previous problems requires an excess of air of 35 per cent over that chemically necessary, to give best results. What is the amount of air required per cubic foot of the gas?

Solution.

Volume of air chemically necessary is

$$1.00 \text{ cubic foot per cubic foot of gas}$$

Volume of air necessary to give best results is

$$1.00 \times 1.35 = 1.35 \text{ cu. ft. per cu. ft. of gas}$$

Heat Value of Gas. The heat value of the gas is determined by multiplying the per cent of each constituent present by the heat value of that constituent—per pound or per cubic foot in accordance with whether the per cent is by weight or volumetric—and dividing the sum of the products so obtained by 100.

Example. What is the low heat value per cubic foot of the producer gas of the previous problems?

Solution.

	Per cent by Volume		Heat Value		
CO_2	= 5.2	×	0	=	0
O_2	= 0.4	×	0	=	0
CO	= 22.0	×	322	=	7373
H_2	= 15.3	×	281	=	4299
CH_4	= 1.0	×	913	=	913
N_2	= 55.2	×	0	=	0
	100.0				12,585

$$\text{Heat value} = \frac{12{,}585}{100} = 125.9 \text{ B.t.u. per cu. ft.}$$

Heat Value of Explosive Mixture. The heat value of the explosive mixture per cubic foot can be determined by dividing the heat value of a cubic foot of the gas by the sum of the volumes of the gas and air.

Example. What is the low heat value of the explosive mixture of the producer gas of the previous problems and air (1) when just the amount of air chemically necessary is present, and (2) when there is an excess of air of 35 per cent over that chemically necessary?

Solution.

(1) Low heat value of the explosive mixture

$$\frac{125.9}{1+1.00}=63.0 \text{ B.t.u. per cu. ft.}$$

(2) Low heat value of the explosive mixture

$$\frac{125.9}{1+1.35}=53.6 \text{ B.t.u. per cu. ft.}$$

Contraction in Volume. The contraction due to combustion is the difference in volume between the original explosive mixture and the products of combustion when the latter have been cooled to the original temperature of the mixture.

The contraction is determined by multiplying the volumetric per cent of each constituent present by the volume contraction of that constituent, as given in Table V, and dividing the sum of the products so obtained by 100.

Example. What is the contraction, water vapor uncondensed, of the producer gas of the previous problem in burning?

Solution.

	Per cent by Volume		
CO_2 =	5.2		
O_2 =	0.4	×1.0 =	0.40
CO =	22.9	×0.5 =	11.45
H_2 =	15.3	×0.5 =	7.65
CH_4 =	1.0	×0.0 =	0.0
N_2 =	55.2		
	100.0		19.70

Contraction $=\frac{19.7}{100}=0.197$ cu. ft. per cu. ft of gas burned with water vapor uncondensed

Volumetric Analysis of Exhaust Gas. The volumetric analysis of the probable exhaust gas can be determined by multiplying the volumetric per cent of each constituent present by the volume of water and carbon dioxide formed by the combustion of that constituent, as given in Table IV; adding the per cent of nitrogen in the gas to the volume of nitrogen present in the air required for the combustion of 100 cubic feet of the gas; determining the volume of oxygen in excess of that chemically necessary for the combustion of 100 cubic feet of the gas; then adding all these quantities, dividing each item by the sum, and multiplying by 100.

Example. What is the probable volumetric analysis of the exhaust gas from the combustion of the producer gas of the previous problems in 35 per cent excess air?

Solution.

Volume of air chemically necessary for combustion

1.00 cu. ft. per cu. ft. of gas

Volume of air required (35 per cent excess)

135 cu. ft. per 100 cu. ft. of gas

Volume of nitrogen in the air

$135\times0.79=106.7$ cu. ft. per 100 cu. ft. of gas

Volume of oxygen in the air

$135-106.7=28.3$ cu. ft. per 100 cu. ft. of gas

Volume of oxygen chemically necessary

$100\times0.21=21.0$ cu. ft. per 100 cu. ft. of gas

Volume of oxygen remaining in the exhaust gas

$28.3-21.0=7.3$ cu. ft. per 100 cu. ft. of gas

Volume of the exhaust gas may be calculated thus:

		H_2O	CO_2	N_2	O_2
	$CO_2=5.2$		$\times1=5.2$		Used in combustion
	$O_2=0.1$				
	$CO=22.9$		$\times1=22.9$		
	$H_2=15.3$	$\times1=15.3$			
	$CH_4=1.0$	$\times2=2.0$	$\times1=1.0$		
	$N_2=55.2$			55.2	
From the air	$N_2=106.7$			106.7	
	$O_2=7.3$				7.3
	Total	17.3 +	29.1 +	161.9 +	7.3 = 215.6

Check.

Volume of air and gas before combustion

$135+100=235$ cu. ft. per 100 cu. ft. of gas

Contraction due to combustion

19.7 cu. ft. per 100 cu. ft. of gas burned

Volume after combustion

$235-19.7$, or 215.7 cu. ft. per 100 cu. ft. of gas

Calculations for Exhaust Gas. In all apparatus for the analysis of gas, the gas is cooled off and the water vapor condensed out, and therefore the analysis of the exhaust gas should be given on a dry basis. In this case it will be given on both.

	Volume from Combustion of 100 cu. ft. Gas.	Analysis: Per Cent of Water Vapor Uncondensed	Analysis: Per Cent of Water Vapor Condensed
H_2O	17.3	$\frac{17.3}{215.6}\times 100 = 8.0$	
CO_2	29.1	$\frac{29.1}{215.6}\times 100 = 13.5$	$\frac{29.1}{198.3}\times 100 = 14.7$
O_2	7.3	$\frac{7.3}{215.6}\times 100 = 3.4$	$\frac{7.3}{198.3}\times 100 = 3.7$
N_2	161.9	$\frac{161.9}{215.6}\times 100 = 75.1$	$\frac{161.9}{198.3}\times 100 = 81.6$
		100.0	100.0

Therefore the volume of exhaust gases per 100 cubic feet of gas is as follows:

With water vapor uncondensed = 215.6 cu. ft.

With water vapor condensed = 198.3 cu. ft.

The *analysis* by *weight*, the *specific weight*, the *density*, and the *specific heats* of the exhaust gas can be determined in precisely the same manner as for the producer gas.

EXAMPLES FOR PRACTICE

1. What is the analysis by weight of the natural gas given in Table VI?

Ans.

H_2, 0.3; CH_4, 90.9; C_2H_4, 0.5; CO_2, 0.5; CO, 1.0; O_2, 0.8; N_2, 6.0

2. What is the specific weight and the density of the gas?

Ans. $\begin{cases} \text{Specific weight} = 0.0431 \text{ lb. per cu. ft.} \\ \text{Density} = 0.566 \end{cases}$

3. What are the specific heats, at constant pressure, and at constant volume?

Ans. $\begin{cases} C_p = 0.567 \\ C_v = 0.441 \end{cases}$

4. What are the weight and volume of air chemically necessary for combustion?

Ans. $\begin{cases} \text{15.95 lb. of air per lb. of gas} \\ \text{8.96 cu. ft. of air per cu. ft. of gas} \end{cases}$

5.* If, to obtain best results, the air must be 56 per cent in excess of that chemically necessary, what is the volume of air actually necessary? Ans. 13.97 cu. ft. of air per cu. ft. of gas

6. What is the low heat value per cubic foot of the gas, and of the ideal and actual explosive mixture?

Ans. $\begin{cases} \text{861 B.t.u. per cu. ft.} \\ \text{86.4 B.t.u. per cu. ft.} \\ \text{57.5 B.t.u. per cu. ft.} \end{cases}$

7. What is the contraction due to combustion?

Ans. 0.013 cu. ft. per cu. ft. of gas burned

8. What is the volumetric analysis of the probable exhaust gas from the actual explosive mixture (1) steam uncondensed and (2) steam condensed?

$$\text{Ans.}\begin{cases} H_2O\ 12.6, & CO_2\ \ 6.3, & N_2\ 74.0, & O_2\ 7.1 \\ & CO_2\ \ 7.2, & N_2\ 84.8, & O_2\ 8.0 \end{cases}$$

9. Check the volume of air chemically necessary for the gases of Table VI.

PROPERTIES OF GASEOUS FUELS

Illuminating Gas. In those regions where *natural gas* occurs, that fuel is used almost exclusively in the gas engine; but in most regions the gas has to be made from either solid or from liquid fuels. The use of liquid fuels will be considered later in connection with the discussion of the oil engine. In most towns of moderate size, there is available illuminating gas made from coal. The illuminating gas is made by one of two processes, giving either *coal gas* or *water gas*. There may also be available *coke-oven gas*, *oil gas*, or other special gaseous fuels.

Coal Gas. Coal gas is made by heating bituminous coal in a retort, away from contact with the air, so that no combustion takes place. The hydrocarbon gases in the coal are driven off by the heat, and, after undergoing various purifying processes, are collected in a holder. The non-volatile part of the coal remains as coke. The gas consists mainly of hydrocarbons, and has a high heating value.

Water Gas. Water gas is made from a non-gaseous fuel, such as anthracite coal or coke, by an intermittent process. Air is blown through a bed of coal several feet thick, until the coal is incandescent, the products of combustion being permitted to escape. Then a jet of steam is blown through the incandescent fuel, and is thereby broken up into its constituent elements—hydrogen and oxygen. The oxygen combines with the carbon of the fuel to form carbon monoxide (CO); the hydrogen goes off unchanged. The passage of the steam quickly cools the coal, and air has to be blown through again. The only gas collected is that generated during the steam blow; it consists principally of hydrogen and carbon monoxide, and has a much lower heating value per cubic foot than coal gas. The whole of the coal is consumed in this process. If this gas is to be used for

TABLE VI

Volumetric Composition, Heat Value, Etc., of Fuel Gases

	Natural Gas	Oil Gas	Coal Gas	Carbureted Water Gas	Coke-Oven Gas	Water Gas (Blue)	Producer Gas		Blast-Furnace Gas
							From Anthracite or Coke	From Bituminous Coal	
	Per Cent	Per Cent	Per Cent	Per Cent	Per Cent	Per Cent	Per Cent	Per Cent	Per Cent
Hydrogen.....................H_2	2.0	58.4	46.0	40.0	48.0	48.0	13.0	10.0	.7
Marsh gas.....................CH_4	93.0	28.8	40.0	25.0	35.5	2.0	.8	3.0	....
Olefiant gas or Ethylene.........C_2H_4	.3	3.4	5.0	8.5	5.4	....	....	.5	....
Carbon dioxide.................CO_2	.2	1.2	.5	3.0	1.3	6.0	4.0	5.0	11.7
Carbon monoxide ....CO	.6	4.4	6.0	19.0	5.1	38.0	27.0	23.0	26.7
Oxygen......................O_2	.4	....	.5	.5	0.5	.5	.5	.5	....
Nitrogen.....................N_2	3.5	3.8	2.0	4.0	4.2	5.5	54.7	58.0	60.9
B.t.u. per standard cu. ft..............	900–1,000	600–700	550–650	550–600	340–645	300–350	130–150	130–150	90–100
Air chemically necessary per cu. ft. of gas	9.0	4.8	5.7	5.0	5.4	2.2	1.0	1.1	.66
B.t.u. per cu. ft. of ideal mixture......	100	110	90.0	96.0	91.0	100.0	67.5	64.0	58.0
Air actually necessary per cu. ft. of gas to give best results..................	14.0	7.0	8.2	7.2	7.8	3.1	1.35	1.5	1.0
B.t.u. per cu. ft. of actual mixture....	67.0	81.0	65.0	70.0	66.0	79.0	57.5	54.0	48.0

illuminating purposes, it has to be enriched by the addition of hydrocarbon vapors obtained by heating crude oil or other oil.

Both coal gas and water gas are excellent fuels for use in a gas engine; but since, for cleansing them and increasing their illuminating power, they have gone through certain processes which increase the cost of the gas, but do not add materially to its value for gas-engine use, and since, also, the cost to the consumer is considerably greater than the cost of production, they are not economical fuels. Such fuels should be used only when the engine is small or its operation infrequent.

Coke-Oven Gas. When bituminous coal is heated in a retort, the products of the process are gas, tar, ammonia, and coke. In city gas plants, the first is the principal product and the rest are by-products. There are in this country a considerable number of by-product coke ovens which carry out the same process as in city gas plants, but with a different purpose in view, and with more complete separation of the by-products. The coke-oven gas, which is obtained as a by-product from such ovens, does not differ materially from coal gas.

Oil Gas. Oil gas is obtained by mixing the vapor of crude or other mineral oil with superheated steam, and sending the mixture to a retort where a temperature of about 600° F is maintained. The vapor is there converted into a non-condensable gas very rich in hydrogen.

Blast-Furnace Gas. Blast-furnace gas is the gas that comes from the top of a blast furnace, which is simply a huge gas-producer. In the past, it has either been burned there, and consequently wasted, or has been burned under boilers for generating steam. It is a much weaker gas than any of the others described, but can be used most satisfactorily and economically in gas engines. Naturally, it is available only at blast-furnace plants.

Producer Gas. If gas is not taken from any of these sources, it can be generated specially for the engine in a *gas-producer*.

In the gas-producer, either air alone, or generally both air and steam, are sent through a thick bed of coal. The oxygen of the air, on first striking the zone of the incandescent coal, combines with the carbon to form carbon dioxide (CO_2); but this, on passing through the burning coal above, is reduced to carbon monoxide (CO), which

escapes with the hydrogen and carbon monoxide resulting from the action of the steam on red-hot coal and with the nitrogen which came in with the air. The resulting gas, therefore, consists almost entirely of carbon monoxide, hydrogen, and nitrogen. The large amount of nitrogen in the air (79 volumes in 100) makes producer gas contain 50 per cent or more of that inert gas, and consequently gives it a low heat value.

Characteristics Compared. *Composition.* The compositions of the various gases mentioned are given in Table VI (p. 53). They are all rich in hydrogen and marsh gas, with the exception of blast-furnace gas and producer gas. The presence of large quantities of hydrogen makes a gas engine peculiarly liable to premature ignition. As this phenomenon is particularly pronounced and particularly objectionable in large gas engines, those gases only which contain not more than 10 to 12 per cent of hydrogen are desirable for large powers.

Heat of Combustion. The heat of combustion of a cubic foot of each of the gases under the standard conditions—that is, with the gas at 62° F. and at a pressure of 14.7 pounds per square inch—is also given in Table VI. There is a very large range in the values, the extreme range from natural gas to blast-furnace gas being a range of 12 to 1.

Volume of Air Used in Combustion. The volume of air chemically necessary for the combustion varies, however, through a range which is almost as great; for natural gas it is 9 times the volume of the gas; for blast-furnace gas only ⅔. The volume of air actually necessary varies through a greater range; for natural gas it is 14 times the volume of the gas; for blast-furnace gas only 1.

Heat of Combustion of Mixture. The heat of combustion of a cubic foot of the perfect explosive mixture is, for natural gas, about 100 B.t.u.; for blast-furnace gas, about 60 B.t.u.; that is, the heat of combustion of a cubic foot of the explosive mixture does not vary much, even in the two extreme cases. In practice, more air goes to the cylinder with the gas than the amount that is chemically necessary; an excess of at least 35 per cent over that amount is usual. Such excess of air results in more complete combustion, and consequently gives greater economy. Table VI gives the average heat of combustion per cubic foot of the theoretical mixture and of the actual mixture.

PROPERTIES OF LIQUID FUELS

Internal-combustion motors can be made to work with any explosive mixture. Mixtures of air with gaseous fuels are naturally the mixtures most easily made and controlled. Generally, mixtures of air with liquid fuels offer no particular difficulty. Mixtures of air with solid fuels, such as powdered coal, have been tried, but are not practicable, on account of the ash which remains in the cylinder and rapidly abrades it.

The liquid fuels which are commercially available are *crude petroleum and its distillates*, and *alcohol*.

Crude Petroleum. Crude petroleum occurs at many parts of the earth's surface, the principal sources being the United States and the Baku district in the Caucasus. In the United States the principal fields are in Pennsylvania, Ohio, Texas, and California. The oils from these different fields are very different in their characteristics. They consist almost entirely of compounds of hydrogen and carbon—the so-called *hydrocarbons*. The crude oils are made up principally of closely related compounds, some of which, on separation, are gaseous, others liquid, and still others solid at ordinary temperatures. The liquid constituents are of different densities and volatilities, varying from an extremely light liquid, which evaporates rapidly at atmospheric temperature (just as alcohol and ether do), to heavy, viscous liquids, which have to be raised to a high temperature before they will give off vapors. The character of the crude oil depends on the relative amounts of these various constituents. The Pennsylvania, Ohio, and Baku oils contain a considerable proportion of the lighter liquid constituents. The Texas and California oils contain very little of the lighter constituents, but consist mainly of a different series of hydrocarbons, having close chemical relations with asphaltum.

The crude oils from Pennsylvania and Ohio can be used in oil engines. The Texas and California crude oils can also be used, but only with difficulty and in engines specially designed for such oils. The crude oil, because it is a mixture of substances of very divergent physical properties, is not a satisfactory fuel; those engine conditions which are favorable for burning one part of the oil are not necessarily favorable for the other constituents.

Refining Products. The Pennsylvania and Ohio crude oils are commonly refined before using. The refining is a process of distillation carried on in closed retorts. If crude petroleum is slowly heated, it gives off as vapor its various constituent elements; the more volatile being given off at the lower temperatures, and the residue becoming continuously more dense and more viscous. In the refining of petroleum, the vapors given off at various temperatures are condensed and collected separately; the names given to the various products are an index chiefly to the temperature at which they give off their vapors. The most volatile of the ordinary products contains all the elements that vaporize at a temperature below 160° F., and is called *gasoline.* It gives off some of its lighter vapors at the ordinary temperature of the air; and, as these vapors are highly combustible, gasoline is quite dangerous. When mixed with from 8 to 20 parts of air, it forms an explosive mixture which gives a more rapid explosion, and consequently higher pressure, than do mixtures of equal heat value with any of the gaseous fuels. When exposed to the air, the lighter vapors escape, leaving behind a heavier and less volatile oil.

If petroleum which has been heated for some time at 160° F. is slowly raised in temperature to 250° F., a new and heavier series of vapors will be given off, which, when condensed and collected, are called *benzine* or *naphtha.* On further raising the temperature from 250° F. to 350° F., a still heavier series of vapors is given off, forming the oil known as *kerosene.* Kerosene will not give off inflammable vapors till it is heated to about 120° F., so that it is comparatively safe, and will not change or deteriorate when stored under ordinary conditions. It is more difficult to burn satisfactorily than is gasoline; and, when subjected to a high temperature with insufficient air for its combustion, it decomposes and deposits its carbon as a hard cake on the walls of the containing vessel. The dense petroleum which remains after the kerosene has been driven off is called *fuel oil.* If the fuel oil is subjected to still higher temperatures, other and denser vapors are driven off, giving, when collected, *lubricating oils*, *cylinder oil*, and *paraffine wax*, and leaving, finally, a dense, sticky mass, which is known as *residuum.*

The ordinary distillation is into three "fractions"; but the distillation can be made in as many steps as desired, and by re-distilla-

TABLE VII

Densities Corresponding to Degrees Baumé for Liquids Lighter Than Water

Degrees Baumé	Density	Degrees Baumé	Density	Degrees Baumé	Density	Degrees Baumé	Density
10	1.0000	30	0.8750	50	0.7778	70	0.7000
12	0.9859	32	0.8642	52	0.7692	72	0.6931
14	0.9722	34	0.8537	54	0.7609	74	0.6863
16	0.9589	36	0.8434	56	0.7527	76	0.6796
18	0.9459	38	0.8333	58	0.7447	78	0.6731
20	0.9333	40	0.8235	60	0.7368	80	0.6667
22	0.9211	42	0.8140	62	0.7292	84	0.6542
24	0.9091	44	0.8046	64	0.7216	88	0.6422
26	0.8974	16	0.7955	66	0.7143	92	0.6306
28	0.8861	18	0.7865	68	0.7071	96	0.6195

tion more and more complete separation of the individual components can be effected. As the practice in distilling varies in the different oil refineries, an endless variety of distillates of petroleum is purchasable.

Density Best Indication of Properties of Product. The best indication of the general physical properties of any petroleum product is found in its density, as each constituent of the petroleum has a different density. The density is not, however, an entirely satisfactory indication, since a mixture of heavier and lighter oils may have the same density as some intermediate oil.

The density of a liquid is the weight of the unit volume of that liquid at 60° F., as compared with the weight of the same volume of water at 60° F. All the liquid fuels are lighter than water. It is the common practice to speak of the density of petroleum products in degrees Baumé. This is an arbitrary scale, with nothing to recommend it. Its relation to true density for liquids lighter than water is given in Table VII.

The density of gasoline varies from .67 to .71; of kerosene, from .75 to .82; of fuel oil, from .82 to .85.

The higher the density, the less the degrees Baumé. The heats of combustion of the various crude oils and their distillates do not vary greatly; they range from 18,000 to 20,000 B.t.u. per pound.

Denatured Alcohol. The action of the United States Government in removing the excise duty from denatured alcohol has made that substance commercially available for use in internal-combustion motors. There are two principal kinds of alcohol: (1)

TABLE VIII

Heat Values and Mixture Proportions for Common Liquid Fuels

	Gasoline	Kerosene	Alcohol 90 Per Cent by Vol.	Crude Oil
Lower heat value—B.t.u. per lb.	20,500	20,300	10,900	18,000
Air chemically necessary per lb. of fuel—in cu. ft.	189	187	101	176
Heat value of ideal mixture—B.t.u. per cu. ft.*	108.5	108.5	108.0	102.0
Air actually necessary per lb. of fuel to give best results—in cu. ft.	300	300	165	305
Heat value of actual mixture—B.t.u. per cu. ft.*	68.5	67.5	66.0	59.0

* In explosive mixtures of liquid-fuel vapors and air, the volume occupied by the fuel vapor, when compared with the volume of air, is so small that it may be neglected without serious error. In the above table the Heat Value of the Mixture, Ideal and Actual, was calculated on the above assumption.

ethyl or *grain alcohol* (C_2H_6O), which can be made from corn, rye, rice, molasses, beets, or potatoes, by a process of fermentation and distillation; and (2) *methyl* or *wood alcohol* (CH_4O), which is obtained from the destructive distillation of wood. Grain alcohol is that which is present in alcoholic beverages; wood alcohol is a virulent poison. Denatured alcohol is grain alcohol which has been rendered unpalatable and unfit for consumption by the addition of wood alcohol and a little benzine or other substance. The common composition of denatured alcohol is 100 volumes of grain alcohol mixed with 10 volumes of wood alcohol and ½ volume of benzine. This substance contains within itself some of the oxygen which is necessary for its combustion. It gives up about 11,800 B.t.u. per pound on burning, which is not much more than one-half as much heat per pound as gasoline or kerosene.

Data on Liquid Fuels. Table VIII gives the lower heat value in B.t.u. per pound; the volume of air, chemically and actually necessary, per pound of fuel; and the heat value of the ideal and actual explosive mixture in B.t.u. per cubic foot, for the more common liquid fuels.

EXPLOSIVE MIXTURES

Proportions of Gas and Air for Various Fuels. An important characteristic of a fuel is its explosibility with an excess or deficiency of air. It is not possible or desirable to regulate the air supply to an engine so that there shall always be present exactly the amount chemically necessary. Other things being equal, that fuel is best which will permit the largest variation of the ratio of air to fuel without failure to ignite. Coal gas, which unites with 5 to 7 times its own volume of air, will ignite—at atmospheric pressure—with any amount of air between 4 and 12 times its own volume; water gas (uncarbureted), using 3.9 times its own volume of air, will ignite between the limits of 0.5 and 7 times its own volume. That is, an engine using uncarbureted water gas will function under a much larger variation of the ratio of air to gas than will a coal-gas engine. To get complete combustion, the air supply must always be somewhat in excess of that chemically necessary. If it is much in excess of that amount, the combustion may be complete, but it will be slower and will not give such good efficiencies.

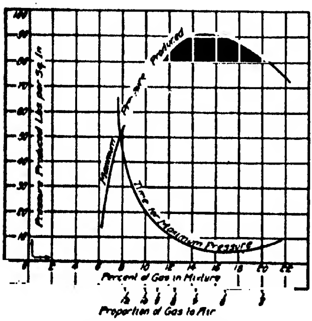


Fig. 14. Pressure Curves from Explosion of Uncompressed Coal-Gas Mixtures

Explosibility of Various Proportions of Coal Gas. The curves, Fig. 14, are for mixtures of coal gas and air at atmospheric pressure exploded in a closed vessel. They show the effect of the ratio of air to gas on the maximum pressure obtained by the explosion, and on the time it takes the mixture to reach its maximum pressure. It is seen that a mixture of 1 part of gas to about 6¾ parts of air gives the maximum pressure 92 pounds, absolute; and also that the same, or a slightly stronger mixture, gives the minimum duration of the explosion, a duration in the neighborhood of .04 of a second. With the weakest mixture, the time required to reach maximum pressure—

TABLE IX

Limits of Proportion for Explosive Air-Gas Mixtures

Gas		Per Cent of Gas in the Mixture by Volume		
		Combining Proportion	When Air is in Excess	When Gas is in Excess
Carbon monoxide		29.6	16.5	74.95
Hydrogen		29.6	9.45	66.4
Methane		9.5	6.1	12.8
Ethane		5.6	4.0	22.0
Ethylene		6.5	4.1	14.6
Acetylene		7.7	3.35	52.3
Pentane		2.6	2.4	4.9
Benzene		2.7	2.65	6.5
Gasoline	86° Baumé		1.54	4.76
	71° Baumé		1.54	4.76
	65° Baumé		1.31	4.76
Alcohol (95.14% by weight)		6.5	3.95	13.65
Ether		3.4	2.75	7.7
Water gas			10.5	52.3
Coal gas			6.7	18.4
Illuminating gas			7.9	19.1
Blue oil gas			4.0	8.0

which is approximately the time required for complete combustion —is about one-half second. As a small gas engine may run at 360 revolutions per minute, or 6 revolutions per second, there is only $\frac{1}{12}$ of a second available for each stroke; and consequently an explosion requiring ½ of a second is altogether inadmissible.

Effect of Compression on the Explosion. The compression of the charge, which takes place in all gas engines, makes the pressure of the explosion much greater, and its duration less, than those shown in Fig. 14. With a compression to 60 pounds of the best mixture of

TABLE X

Limits of Proportion for Explosive Air-Gas Mixtures at Different Temperatures

Mixture	Temp 59° F		212° F.		392° F.		572° F.	
	Upper Limit	Lower Limit	Upper Limit	Lower Limit	Upper Limit	Lower Limit	Upper Limit	Lower Limit
Hydrogen and air	64.7	9.5	68.2	9.5	72.1	9.6	79.3	9.6
Carbon monoxide and air	74.6	14.3	77.2	13.2	80.4	12.5	57.4	21.0
Methane and air	13.0	6.8	12.6	5.8	12.8	5.8	13.0	5.7
Illuminating gas and air	22.6	7.0	24.7	7.0	26.7	6.5	28.6	6.5

coal gas and air, the explosion in a small engine may be complete in about .01 of a second. With gasoline, the time is even shorter. Table IX gives the limiting proportions, or percentages, of gas in explosive mixtures, between which the mixture is combustible, for the various fuels and constituent gases. Table X gives the limits for mixtures of combustible gas and air for four temperatures, between 59 and 572 degrees F., for the more common of the constituent gases and for illuminating gas.

FUEL-MIXING DEVICES

Process of Carburetion. In order to make an explosive mixture of a liquid fuel with air, it is necessary first to convert the liquid fuel into a vapor or gas. The lighter distillates—gasoline, naphtha, etc., are easily vaporized; the illuminating oils offer some difficulty; the fuel or crude oils are still more difficult.

The cycle of operations through which the engine goes, and the general structure of the engine, may be the same for all these oils as for the gas engines already discussed; the only essential difference is in the addition of devices for supplying the oil to the cylinder, and for its preparatory treatment.

With the lighter oils, the apparatus for vaporizing the oil is called a carbureter; with the heavier oils, a vaporizer.

The vaporization of gasoline is effected by bringing the current of air that is on its way to the cylinder, over, through, or in some other way, into intimate contact with the gasoline. A given volume of air will take up an amount of gasoline which depends on the composition of the gasoline, the temperature of the air and gasoline, and the humidity of the air. When it has taken up its charge of gasoline vapor, the air is said to be "carbureted". The lighter (and more volatile) the gasoline, the more of it will be vaporized by a given volume of air; the higher the temperature of the air and gasoline, the more gasoline is evaporated; also, the drier the air, the greater is its capacity for taking up the gasoline.

Avoiding Selective Evaporation. Gasoline is a mixture of many components; and on the passage of air over a surface of gasoline, the more volatile components vaporize first, leaving a residue, which becomes denser and denser and which gives off vapor at a constantly decreasing rate. As it is desired that all of the gasoline should be

TABLE XI

Fractional Distillates for Gasoline and Kerosene and Temperatures at Which They Are Given Off

Tenths	Temperature Limits in Degrees F.	Specific Gravity of the Distillate	Tenths	Temperature Limits in Degrees F	Specific Gravity of the Distillate
Gasoline (Specific Gravity = 0.684)			Kerosene (Specific Gravity = 0.801)		
1	108–140	0.648	1	280–350	0.755
2	140–145	0.655	2	350–387	0.765
3	145–154	0.665	3	387–414	0.776
4	154–160	0.670	4	414–457	0.783
5	160–167	0.675	5	457–487	0.796
6	167–181	0.686	6	487–525	0.805
7	181–190	0.693			
8	190–205	0.704			
9	205–223	0.718			

used in all engines, and that there should be no variation in the composition of the carbureted air, this selective evaporation has to be prevented. The process of vaporization of the gasoline necessitates the supply of the latent heat of vaporization partly from the air and partly from the body of the fluid. This results in a cooling of the gasoline, which in turn diminishes the rate of vaporization. In cases where the arrangements are such that this cooling of the body of the oil by vaporization is possible, it is necessary to supply heat from outside—either from the exhaust gases or the jacket water—to make up for the loss of heat. The necessary amount of heat for the vaporization of gasoline is small and can be taken from the jacket water.

It is not necessary, however, in all cases to supply heat from outside to the carbureter. There are many devices by which selective evaporation and the cooling of the body of the gasoline can be entirely prevented, but even with such devices it is sometimes desirable to raise the temperature of the gasoline somewhat, so as to increase its volatility.

In Table XI are shown average fractional distillates for gasoline and kerosene, and the temperatures at which they are given off. The column headed "Tenths" refers to tenths by volume of the original fuel evaporated between the temperature limits given in the next column.

Air which at ordinary temperatures has passed over or through the ordinary gasoline of commerce, and is consequently saturated with the vapor of the gasoline—that is, contains as much gasoline vapor as it is possible for it to carry—is too rich in fuel to be explosive. If the temperatures are low or the gasoline dense, this may not be the case. It is necessary, with such rich mixtures, to add more air to the carbureted air in order to get an explosion in the cylinder; or, at any rate, even if an explosion is possible, in order to get an economical performance of the engine. Such admixture of air with the carbureted air may take place either at the cylinder or in the carbureter itself.

CARBURETERS

Types for Automobile and Motor-Boat Work

Classification. The carbureters used for automobiles or motor boats may be divided into three classes, according to the method by which the air and gasoline are brought into contact.

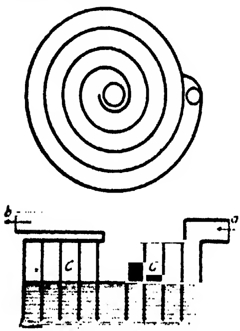


Fig. 15 Surface Carbureter

Surface Carbureters. In the surface type of carbureter, air is made to pass over a gasoline surface, or an extended surface wetted with gasoline. The most simple form is a wick or flannel carbureter, such as is shown in Fig. 15. The air, entering at *a*, is forced by the sheet-metal spiral *c c* to pursue a spiral path till it gets to the center of the carbureter, when it escapes from *b*. The metal spiral has flannel on its surface, and the whole vessel is half-full of gasoline. The air, passing through the carbureter, comes in contact with an extended gasoline and gasoline-wetted surface, and is thereby saturated with vapor The objections to this type are: (1) selective evaporation and (2) cooling of the mass by the vaporization.

Bubbling Carbureters. In the bubbling type of carbureter, air is made to pass through a moderate depth of gasoline, and, in bubbling

through it, becomes saturated. The same objection holds as with the surface carbureter. Both these types are now superseded by the third type.

Spray Carbureters. With the spray type of carbureter the amount of gasoline required for carbureting during one admission to the cylinder is sprayed into the entering air, being thereby partly vaporized and partly atomized, and consequently is carried into the cylinder partly as a vapor, partly as a liquid. In consequence of its separation from the main body of the gasoline, there is no cooling action on the mass of the gasoline by the vaporization, and no alteration in its composition by selective evaporation. If heating of the main body of the gasoline is used, it is in order to increase its volatility and not to make up for cooling by vaporization.

The spraying of the gasoline must occur only when air is being drawn into the cylinder; consequently, it is possible and usual to make the spraying result from the action of the suction during the admission stroke.

Schebler Model "D" Carbureter. In the most common forms of carbureter the gasoline is kept at a constant level by means of a float. In Fig. 16 when the U-shaped float *F*—which is hinged at *J*—falls, it lifts the needle valve *H*, permitting gasoline to enter by gravity from the reservoir, through *G*, into the float chamber *B*. As the gasoline rises in the chamber, it lifts the cork float *F* and closes the gasoline-admission valve. The float consequently keeps the gasoline at a constant level. This constant level is a little below the outlet of the spraying nozzle *D*. Air enters the carbureter on each suction stroke of the engine; and, passing through the mixing chamber *C* with considerable velocity, creates a slight vacuum there, sufficient to suck gasoline up through the spraying nozzle and to cause an intimate mixture of the gasoline with the air. The amount of gasoline admitted is controlled by the needle valve *E*.

Air Supply Adjustment. This carbureter has another feature in common with most automobile carbureters—namely, a device for automatically adjusting the opening for the air supply as the engine speed changes. The compensating air valve *A* remains in the position shown when the engine is going at its lowest speed. As the speed increases, the velocity of the air through the carbureter is greater, and consequently the vacuum in the mixing chamber is

increased. This results in an opening of the valve A against the resistance of the spring O, to an extent which depends on the engine speed. At the same time, the increased vacuum in C increases the gasoline flow through D. The increase of both air and gasoline should be in the same proportion so that, when once adjusted for a good mixture, the variation in speed of the engine should not alter that mixture. The adjusting screw M varies the tension on the

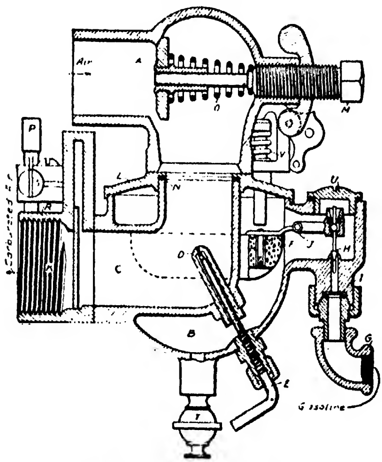


Fig. 16. Schebler Carbureter, Model "D"
Courtesy of Wheeler and Schebler, Indianapolis, Indiana

spring O, and by this means the amount of air valve A opening can be regulated. The throttle valve K, worked by the lever P, controls the amount of the carbureted air going to the engine. The flushing pin or tickler V, when pushed down, keeps the float depressed and permits gasoline to flow through D into the mixing chamber C before starting, so as to insure the admission of an explosive mixture to the cylinder when starting up.

This type of carbureter is used on constant-speed engines, such as two-cycle marine engines, single- and two-cylinder, two- and

four-cycle farm engines, where extremely low throttling is not required.

Schebler Model "L" Carbureter. The Schebler carbureter shown in Fig. 17 is used for automobile and other engines where low throttling and extremely variable speeds are required. In this carbureter independent adjustments of the gasoline feed for the idling, the half-open, and the full-open positions of the throttle are provided. The air passage *K* is in the form of a Venturi nozzle with the gasoline nozzle *L* located at the throat, thus insuring maximum

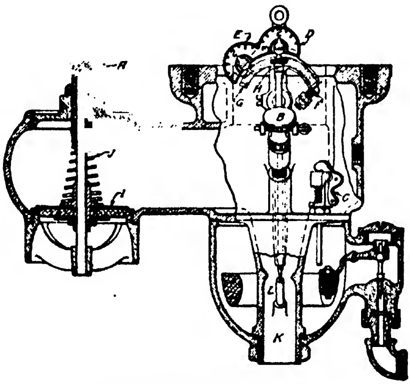


Fig. 17 Schebler Carbureter, Model "L"
Courtesy of Wheeler and Schebler, Indianapolis, Indiana

velocity of the air and thorough carburetion. For automobile engines, the auxiliary air valve *I* can be provided with a dash control, so that the amount of auxiliary air can be regulated from the seat to facilitate starting and to adjust for atmospheric changes. The auxiliary air valve *I*, the gasoline needle valve *B*, and throttle lever screw *F* are adjusted at low speed, as in the case of Model "D". The dials and screws *D* and *E* give the intermediate and high-speed adjustments by raising or lowering the height of the spring track *G* at the corresponding throttle-opening positions.

The needle valve B is thus raised more or less by its roller H, which climbs the spring cam track G as the throttle is opened.

Schebler Model "R" Carbureter. The carbureter, shown in Fig. 18 is designed with an adjustment for low speed. As the speed of the motor increases, the auxiliary air valve opens and raises the gasoline needle, thus automatically increasing the amount of fuel. This carbureter has two adjustments—the low-speed needle adjustment, which is made by turning the air-valve cap A, and an adjustment on the air-valve spring for changing its tension by means of the screw F. In this carbureter, as in the previous type, the gasoline nozzle is located at the throat of a Venturi nozzle.

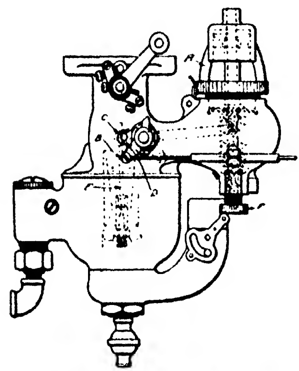


Fig. 18 Schebler Carbureter, Model "R"
Courtesy of Wheeler and Schebler, Indianapolis, Indiana

Holley Model "H" Carbureter. The Holley carbureter, Fig. 19, has some special features. In this carbureter the fuel enters the float chamber through a strainer disk A and a float valve B, under the action of the cork float C. It passes from the float chamber D into the nozzle well E through a passage F. It then enters the nozzle G through the hole H, and rises past the needle valve I, to a level which just submerges the lower end of a small tube J, which has its outlet at the edge of the throttle disk.

Throttling Device. Cranking the engine, with the throttle kept nearly closed, causes a flow of air through the tube J and its throttling plug K. But, as the lower end of this tube is submerged in fuel with the engine at rest, the act of cranking primes the motor. With the motor turning over under its own power, flow through the tube J takes place at high velocity, thus causing the fuel entering the tube with the air to be thoroughly atomized upon its exit from the small

opening at the throttle edge. This tube is called the "low-speed tube" because, for starting and for idle running, all of the fuel and most of the air in the working mixture are taken through it.

As the throttle opening is increased, a considerable volume of air moves through the passage bounded by the conical walls *L* of the so-called strangling tube. In its passage into the strangling tube, the air assumes an annular, converging-stream form, and attains its highest velocity in the immediate neighborhood of the upper end of the standpipe *M*, set on to the body of the nozzle piece *G*. The pressure in the air stream is consequently lowest at the same point, and there is a pressure difference between the top and bottom openings of the pipe *M*, causing air to flow through it from bottom to top.

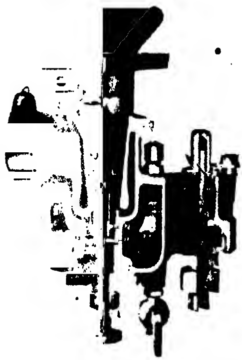


Fig. 19 Holley Carburetor, Model "H"
Courtesy of Holley Brothers Company, Detroit, Michigan

With very small throttle opening, the action through the standpipe (air passing downward through the series of openings *N* in the standpipe supporting bridge) keeps the nozzle cup cleaned out, the fuel passing directly from the needle opening into the entrance of the standpipe.

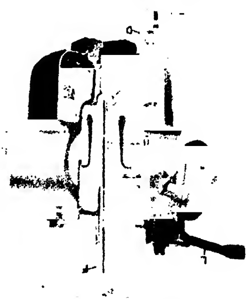


Fig 20. Holley Carburetor, Model "G"
Courtesy of Holley Brothers Company, Detroit, Michigan

Holley Model "G" Carbureter. The mode of operation of the carbureter shown in Fig. 20 is identical with that of Fig. 19. Its chief differences are structural ones.

From the float chamber the gasoline passes through the ports *E* to the nozzle orifice, which is controlled by the end of the needle *F*.

The float level is so set that the gasoline line rises past the needle valve *F* and sufficiently fills the cup *G* to submerge the lower end of the small tube *H*. Drilled passages in the casting connect the upper end of this tube with an outlet at the edge of the throttle disk.

Fuel issuing past the needle valve *F* is immediately picked up by the main air stream at the point of the latter's highest velocity, giving thorough atomization of the fuel.

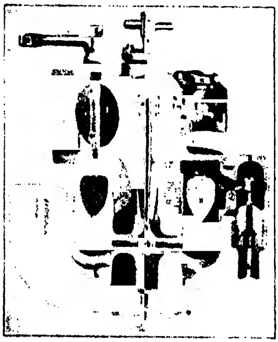


Fig. 21 Kingston Carbureter, Floating-Ball Type
Courtesy of Byrne, Kingston, and Company, Kokomo, Indiana

The lever *L* operates the throttle in the mixture outlet, and a larger disk, with its lever *S*, is a spring-returned strangler valve in the air intake, for facilitating starting in extremely cold weather.

Kingston Floating-Ball Carbureter. The carbureter shown in Fig. 21 differs from that of Fig. 19, in that the fuel nozzle *J* forms a cup, from which the fuel is picked up as the air passes around the nozzle on its way through the Venturi tube, and while the air has its greatest velocity. Another point of difference is that auxiliary air is admitted to the carbureted air through ports in the mixing chamber, which are covered by the bronze balls *L*, and which are opened by the suction of the engine in correct proportion as the engine speed varies, floating at high speeds.

Kingston Model "Y" Carbureter. In the carbureter of Fig. 22 the gasoline is evaporated by causing the air stream to impinge sharply on a well of fuel, the proportions of the mixture being governed automatically by causing the volume of fuel in this well to increase or diminish as the velocity of the air stream is less or greater.

Air enters at the choke throttle and passes down and up through the U-shaped mixing tube. Gasoline enters from the float chamber through an orifice in the bottom of the well controlled by the adjustable needle valve. The normal gasoline level is slightly below the

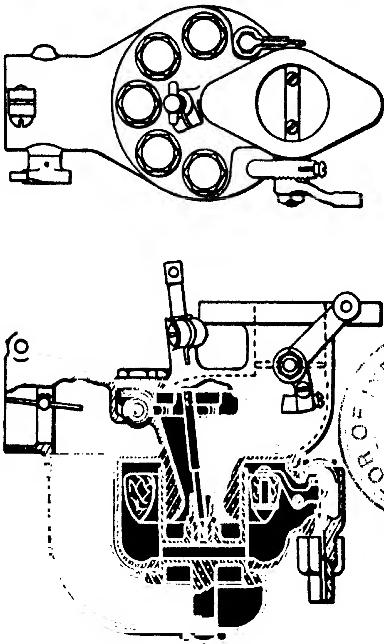


Fig. 23. Kingston Carburetor, Model "Y"
Courtesy of Byrne, Kingston, and Company, Kokomo, Indiana

top of this well. At this point the air passage is constricted, thereby increasing the velocity of the stream. As the motor speed increases, the volume of fuel in the well is gradually diminished, thus preventing the formation of an over-rich mixture. At the highest speeds

the well is wiped completely dry, and an ordinary spray takes its place. Some of the air admitted at the air inlet does not pass over the well, but is by-passed through ports—covered by bronze balls as in the previous carbureter—into the carbureted air. To start, the choke throttle is closed, and the increased suction lifts the gasoline out of the well, thus priming the engine.

Stromberg Carbureters. The carbureters shown in Figs. 23, 24, and 25 differ little in general details from those already described,

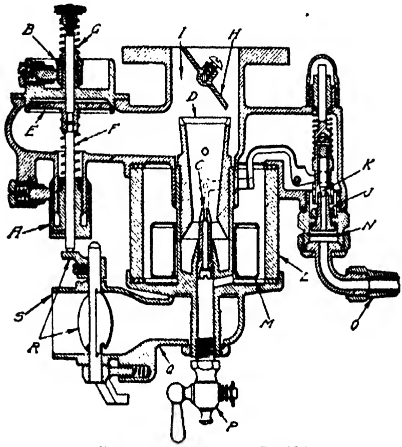


Fig. 23. Stromberg Carbureter, Type "B"
Courtesy of Stromberg Motor Devices Company, Chicago

the chief difference being that, instead of a needle-valve-controlled spray nozzle, they are equipped with a nozzle of fixed opening, and the feed can be adjusted only by removing the nozzle and substituting another with a different opening. The opening of the auxiliary air valve *E* is resisted by two springs. One of these, *G*, is not in tension when the valve is closed—in fact, there must always be at least $\frac{1}{32}$ inch between the spring and the spring washer while the engine is running light. Only a single spring, therefore, is operating during the first part of the opening of the valve while the

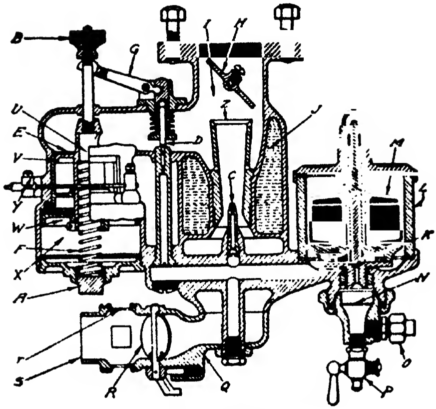


Fig. 24. Stromberg Carbureter, Type "D"
Courtesy of Stromberg Motor Devices Company, Chicago

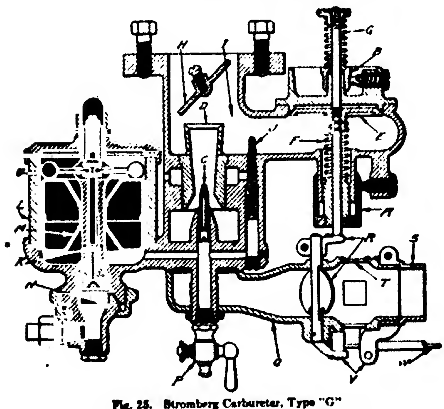


Fig. 25. Stromberg Carbureter, Type "G"
Courtesy of Stromberg Motor Devices Company, Chicago

engine is running at low speed and the suction is low. When the engine speeds up, however, the suction is increased, the valve opening is increased, and the second spring comes into action and resists the further opening of the valve. By adjusting the tension on these springs the carbureters will operate equally well throughout the range of speed of the engine. In the carbureter of Fig. 24, the Venturi tube is jacketed by the hot water from the engine jackets to assist in the vaporization. Fig. 23 shows a single-nozzle carbureter, while the carbureters of Figs. 24 and 25 have double jets. The primary nozzle is the same as in Fig. 23, while the auxiliary nozzle is located between the auxiliary air valve and the mixing chamber. In Fig. 24, the flow from the auxiliary nozzle is regulated by a needle valve, the amount of opening being determined by the amount of opening of the auxiliary air valve. In Fig. 25 the nozzle is a plain spray nozzle with a fixed opening, which, as the feed is regulated by the suction, must be changed to adjust the auxiliary feed.

Types for Slow-Speed Stationary Engines

General Characteristics. The carbureters generally used in the relatively large and slow-speed stationary engines are quite different from those practically in universal use in small high-speed automobile and motor-boat engines. In the latter case, compactness, simplicity, and the absence of a gasoline pump (an appliance not easy to keep tight) are secured. The same type of carbureter, however, is not well adapted to the stationary engine, where larger volumes of carbureted air are required at longer intervals instead of small volumes at short intervals.

Fig. 26. Nash Carbureter
Courtesy of National Meter Company, New York City

Nash Carbureter. The carbureter shown in Fig. 26 is a simple rigid cast-iron device, cylindrical in form, with a water jacket. The

gasoline is pumped to the top of the carbureter, falls over baffle plates, and is partly vaporized. Air drawn through the car-

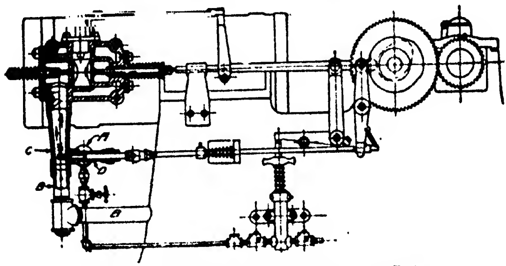


Fig. 27. Details of Fuel Injector, Charter Gasoline Engine
Courtesy of Charter Gas Engine Company, Sterling, Illinois

bureter by the suction in the cylinder meets gasoline vapor and is saturated by it. The carbureted air goes from the top of the carbureter to the engine, while any unvaporized gasoline drains to

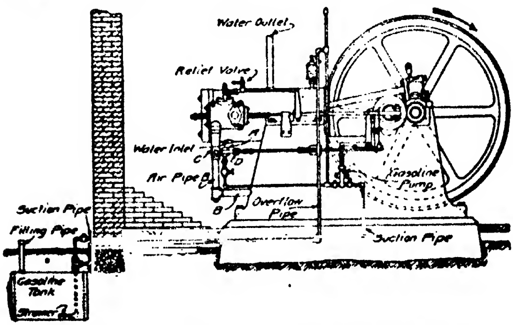


Fig. 28. Arrangement of Fuel Tank and Connections, Charter Gasoline Engine
Courtesy of Charter Gas Engine Company, Sterling, Illinois

the suction side of the gasoline pump and is returned later to the carbureter. The water jacket has circulating through it some of the heated jacket-water from the cylinder.

The spray method of carbureting air is sometimes used in stationary engines, although the carbureter itself is usually of a somewhat different type from the Schebler, Holley, Kingston and others which have been already described in detail on pages 65 to 74. A typical arrangement is illustrated in Figs. 27 and 28, which show the whole arrangement of a gasoline plant and details of the fuel injector. The gasoline tank is buried below the floor level and outside the building, in order to reduce the danger in case of fire or explosion, and also to prevent the leakage of gasoline from the pipes when the engine is not running. The gasoline is taken through a strainer near the bottom of the tank and through the suction pipe by the action of a gasoline pump, which is worked from the camshaft. It is then forced through the control valve *A*, and is sprayed into the air pipe *B* through the jet *C* whenever the fuel-admission valve *D* opens. A vertical branch of the discharge pipe from the gasoline pump has an overflow connecting with the tank. The pump always delivers more gasoline than is required, the excess being returned to the tank through the overflow pipe. This maintains a constant pressure of the gasoline, depending only on the constant overflow level. With a given opening of the control valve *A*, and a constant head on the gasoline, the amount of gasoline admitted each time remains constant.

Fig. 29. Fuel Reservoir for Fairbanks-Morse Type "T" Vertical Engine
Courtesy of Fairbanks, Morse and Company, Chicago

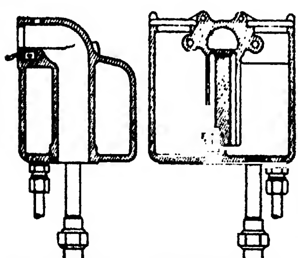


Fig. 30. Sections of Combined Fuel Reservoir on Fairbanks-Morse Type "T" Engine, Showing Compartments for Gasoline (with Check Valve) and Kerosene

Fairbanks-Morse Model "T" Carbureter. Another carbureter, or vaporizer, which is used on stationary and portable farm engines, is shown in Figs. 29 and 30. The fuel—kerosene, gasoline, naph-

tha, or benzine—is supplied to the reservoir by a pump, driven from the camshaft of the engine, and is maintained at a constant level in this reservoir by means of an overflow weir, whose discharge is piped to the fuel tank. The fuel nozzle leading from the reservoir to the air-inlet pipe, is above the level of the fuel in the reservoir, so that the feeding is accomplished by the suction of the engine, and there is no possibility of fuel flowing when the engine is shut down. The opening of the nozzle is hand-regulated by a needle valve, and is also governor-regulated by the mixture valve on the discharge side of the nozzle to keep it correct throughout the range of load.

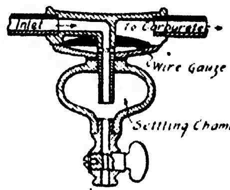


Fig. 31. Gasoline Strainer

Great care must be taken, by the use of suitable strainers, that no solid foreign matter gets into the oil supply pipe; otherwise there is great liability to obstruction of the flow. A strainer for an automobile engine is shown in Fig. 31. Owing to its more rapid explosion, and to the greater richness of the explosive charge, a gasoline engine will develop more power than a gas engine of the same size, even when the latter uses natural gas.

VAPORIZERS

Type for Denatured Alcohol

Volatility and Fuel Value of Denatured Alcohol. The carbureters described in the preceding pages can be used only for the more volatile liquid fuels—liquid fuels with a low boiling point—such as gasoline, naphtha, benzine, etc. The less volatile liquid fuels—liquid fuels with a high boiling point—must be vaporized at a higher temperature by the addition of heat before or during their mixture with air. Denatured alcohol is intermediate between gasoline and kerosene in its volatility. The amount of vapor which it gives off to air that passes over it will generally be sufficient to give an explosive mixture if the temperatures of the air and alcohol are above 70° F. With an ordinary spray carbureter, a considerable excess of alcohol may be sent to the cylinder, as such carbureters act also as

atomizers. If alcohol is supplied in considerable excess, there may still be good explosions, as the range of explosibility is very great. Most of the ordinary gasoline spray carbureters can be used for alcohol if the spray orifices are enlarged. The weight and volume of denatured alcohol required to develop a given power in an engine, is considerably greater than the amount of gasoline for the same power; and therefore, if a gasoline engine is to be used with alcohol, the orifices in the carbureter or other spraying devices have to be enlarged so as to admit a greater volume of the liquid. Wood alcohol cannot be used by itself in a gas engine, as it corrodes the cylinder.

Special Alcohol Vaporizer. A special vaporizer for alcohol is shown in Fig. 32. The hot exhaust gases enter at the bottom, and a certain proportion of them, as determined by the regulating valve, rise to the top of the internal pipe, and then descend between that pipe and the helical cast-iron vaporizer. The alcohol is admitted near the bottom on the outside of the helix and, being vaporized by the heat, flows upward around the helix, escaping to the motor at the top in a highly superheated state. The superheating prevents any condensation of the alcohol between the vaporizer and the cylinder. Air enters with the alcohol vapor as indicated. This vaporizer is of the boiling type, the rate of boiling being determined by the volume of the exhaust gases admitted to the helix.

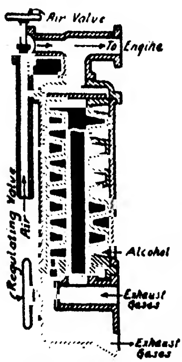


Fig. 32. Alcohol Vaporizer

Recent tests have demonstrated that any gasoline or kerosene engine can operate with alcohol without any structural changes, and that about 1.8 times as much alcohol as gasoline is required to develop the same power. Alcohol can be used with greater compression, as there is little danger of pre-ignition through too much compression, on account of its comparatively high ignition temperature, and also because it is always mixed with some water. An alcohol engine can be made to give somewhat higher power than a

gasoline engine of the same size. It is not so sensitive to maladjustment of the explosive mixture; that is, it will work with a great range of strength of mixture, and it does not accumulate a deposit of carbon inside the engine. A small engine of good design should use about 1.15 pounds of alcohol per brake horsepower per hour; of gasoline, 0.7 pound.

Types for Kerosene and Heavier Fuels

General Method of Vaporization. There are two ways of preparing heavier oils, such as kerosene, crude oil, or fuel oil, for combustion in an engine: (1) by preliminary vaporization; (2) by spraying the liquid in an atomized condition into a cylinder containing compressed air in a high state of compression and at a high temperature, as in the Diesel engine. If a vaporizer is used, it is heated either by exhaust gases on the outside, or by the explosion taking place within it. It requires always a preliminary heating before the engine can be started—unless the engine is started with gasoline—and consequently is not so quickly put in action as a gasoline engine. The vaporization of the heavier oil differs from that of gasoline in that it is not necessarily a process of carburetion. It is often a process of boiling, the mixing with the air required for combustion being subsequent to the vaporization. In other vaporizers the oil is dropped upon a hot plate at the desired rate, and its vapor is carried off by a current of air passing over the plate on its way to the engine.

The principal difficulty with all the vaporizers of the hot-plate type is in keeping the temperature of the plate within the proper limits. If the plate is too hot, the oil decomposes and leaves a deposit of carbon; if it is not hot enough, the vaporization is incomplete.

Vaporizers may be classified as *external* and *internal*, according as the vaporization occurs outside the engine proper or inside some part of the combustion space.

External Vaporizers. Kerosene is sometimes broken up into a fine spray by a current of air, which may be heated by the hot exhaust gases before being carbureted, and is then sent to a vaporizer before being admitted to the cylinder. In the vaporizer the carbureted air is raised to a high temperature, the heat of the exhaust gases being utilized for this purpose, and the kerosene is converted

into a vapor. Unless the kerosene is completely vaporized before admission to the cylinder, it is difficult to insure its complete combustion. Some of the liquid kerosene in the cylinder may decompose or break up into its elements as a result of the very high temperature

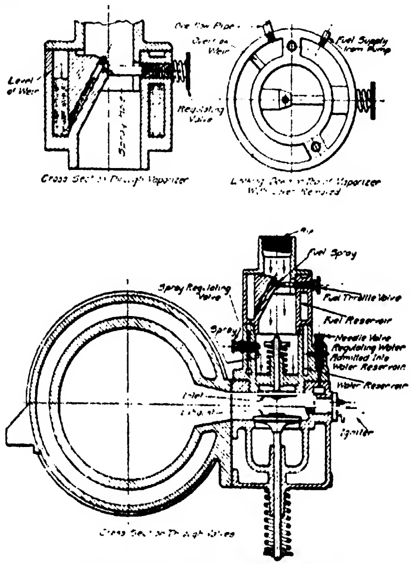


Fig. 33 Details of Vaporizer of Charter Oil Engine
Courtesy of Charter Gas Engine Company, Sterling, Illinois

to which it is subjected, and carbon will then deposit itself on the piston and the walls of the clearance space as a hard coating.

Charter Type "R" Vaporizer for Kerosene and Distillates. In Fig. 33 a device for the vaporization of kerosene and distillates is shown. In this device the air is heated by being drawn through a drum surrounding the exhaust manifold, and is drawn past the vaporizer nozzle. The fuel in thé reservoir is kept at a constant

level, which is lower than the vaporizer nozzle, by means of an overflow weir, so that the spraying is accomplished by means of the suction of the engine. The mixture proportions are regulated by a hand-adjusted valve—seating against the end of the spray hole—the flat end of which, it is claimed, aids in breaking up the fuel into fine particles. Immediately above the inlet valve is located a nozzle for the injection of hot water from the engine jacket into the mixture, the amount of water injection being regulated by a hand-operated valve. This water injection reduces the amount of carbon deposits in the engine and permits of a higher compression being used without pre-ignition.

Unless the vaporization is perfect before the combustion starts, the unvaporized portion of the oil is broken down by the heat and deposits carbon in the engine. In vaporizing, the heat should always

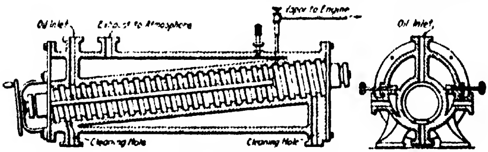


Fig. 34. External Vaporizer for Crude Oil

be high enough to vaporize all the oil, but never high enough to decompose it. If steam or water vapor is present during the combustion the heat may be kept low enough to prevent the decomposition of the oil, due to the heat absorbed by the superheating and dissociation of the water vapor into its elements, hydrogen and oxygen. These elements combine again when the temperature has fallen sufficiently during expansion to permit of it and give back the heat absorbed.

Vaporizers for Crude Oil. Another example of the external vaporizer is shown in Fig. 34, as used for California crude oil. The hot exhaust gases circulate outside the inclined vaporizer; crude oil is admitted at the lower end, and the vapor is taken away from the same end. A revolving cleaner permits the removal, during operation, of the accumulated deposit.

In another vaporizer, Fig. 35, the exhaust from the engine entering at *A* heats up a stationary drum and goes off through a

pipe B. The oil to be vaporized is fed from a pipe F into the channels or buckets of the rotating generating drum C. This drum is driven by the engine at a speed of about one-half revolution per minute.

The drum C is heated by radiation from the stationary drum and rotates so as to carry the fresh oil on to and over the top of the drum, where the more volatile parts are driven off. The vapors pass around the central exhaust pipe B, and are superheated by it on their way to the engine through the pipe D. The air required for combustion enters at E, and is heated and mixed with the vapors. The unvaporized part of the oil drops, as the drum rotates, into the

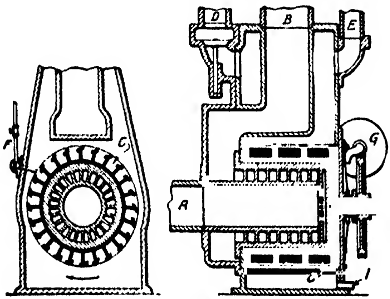


Fig. 35. Crude-Oil Vaporizer

reservoir at the base of the vaporizer, and is automatically drained. With this kind of vaporizer there is little chance of decomposition of the oil by reason of high temperatures; on the other hand, a considerable proportion of a crude oil will go off unused.

Internal Vaporizers. *Fairbanks-Morse Kerosene Atomizer.* The internal vaporizer is always a part of the combustion space of the engine. The device shown in Fig. 36 is a kerosene atomizer used in connection with two-cycle marine engines, and is attached to the by-pass leading from the crankcase to the inlet port of the cylinder. A float chamber is attached to the by-pass at its head, with a nozzle tip entering the by-pass at E. The by-pass A is so shaped as to give the effect of a nozzle with its smallest area at B, so that the velocity

of the air passing from the crankcase to the combustion chamber is a maximum at that point. This increase in velocity causes a decrease in the pressure of the air at B as compared with the air at C. This difference in pressure is used for injecting the fuel. At the point K there is a passage between the upper part of the float chamber and the by-pass, which serves to keep the air pressure in the float chamber the same as that at C. When air is going from the crankcase to the cylinder, the pressure in the float chamber is greater than at B and is sufficient to force a jet of fuel through the nozzle E. The nozzle is placed at right angles to the onrushing air, so that the fuel is broken up or atomized, entering the combustion chamber as a fine spray. The mixture strikes a hot baffle on the piston and is there vaporized. The most common internal

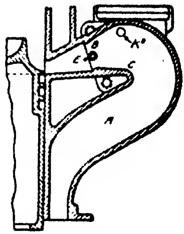


Fig. 36. Section of Fairbanks-Morse Kerosene Atomizer

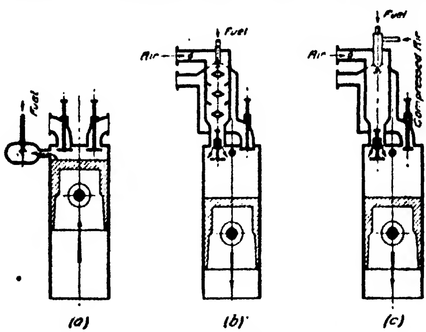


Fig. 37. Methods for Securing Mixture of Vapor and Air

vaporizer is that shown in Fig. 37-a. A combustion chamber or vaporizer is attached to the end of the cylinder, and communicates with it through a narrow neck. The outer part of the vaporizer is unjacketed, and consequently is kept at a good red heat by the successive

explosions. The engine follows the usual four-stroke cycle. During the admission stroke, air alone is admitted to the cylinder, while oil is injected into the vaporizer and is vaporized there. During the return stroke, the air is compressed into the vaporizer, mixes with the oil vapor, and forms an explosive mixture which is ignited by the combined effect of the heat due to compression and the hot walls of the combustion chamber. The proportions of the combustion chamber are designed so that the explosion does not occur until near the end of the compression stroke. The fuel supply is regulated by the governor, which controls a by-pass permitting part of the discharge from the pump to return to the suction side. Before starting the engine, the combustion chamber must be raised to a bright red heat by an external heater; but, after starting, it is maintained in that condition by the explosions. The engine is of great simplicity, since it dispenses with both igniter and mixing valve. The combustion chamber becomes coated with a deposit of carbon, resulting from the break-up of the oil at the high temperature.

*Vaporizer for Use on Regular Gasoline Engines.** An arrangement commonly used by gasoline-engine manufacturers to adapt their engines to the utilization of heavier hydrocarbons is shown in Fig. 37-b. The vaporizer chamber is provided with a jacket space through which the exhaust gases pass, thus heating the vaporizer externally. A cloud of fuel vapor is produced by dropping the liquid fuel on the heated surfaces of the baffle plates inside the vaporizer. Free air enters this vaporizer on the suction stroke of the piston and, in passing over the baffle plates, becomes heated and at the same time absorbs the oil vapors. The mixture thus formed and pre-heated then enters the cylinder and, at the end of the compression stroke, is ignited by an electric igniter.

In Fig. 37-c the fuel oil is mixed with and broken up by a stream of compressed air of from 8 to 25 pounds pressure above atmosphere, so that it enters the vaporizer chamber in the form of finely divided spray and is immediately vaporized due to the heat applied externally by the exhaust gases. The bulk of air, being aspirated during the suction stroke, then mixes with the fuel vapor and becomes pre-heated, thus forming the explosive charge. Compression and ignition are the same as in Fig. 37-b.

*H. R. Setz, *Journal A.S.M.E.*, October, 1911.

Mietz and Weiss Vaporizer. Another form of internal vaporizer for use with any of the heavier oils is shown in Fig. 38. It is applied to a two-cycle engine, and has the further peculiarity that the water in the jacket is permitted to boil, the steam that is formed being taken in with the charge. The oil is taken from the reservoir *A* and pumped through the pipe *B* on to the projecting lip of the hot bulb *C* during the compression stroke. The bulb is heated to a

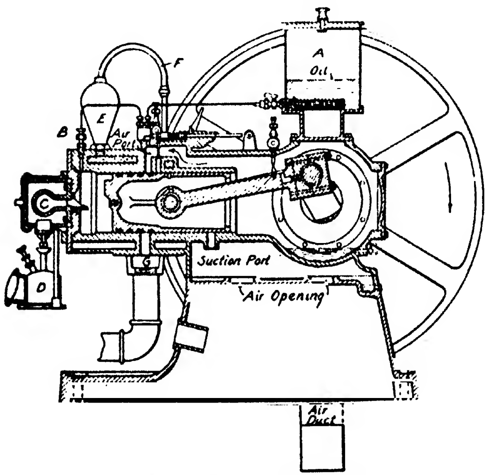


Fig. 38. Section of Mietz and Weiss Oil Engine

dull red heat by the kerosene burner *D* before starting up, and is maintained at that temperature by the explosions when the engine is running. The cylinder head is not jacketed. The amount of oil delivered is regulated by the governor. The air being compressed enters the hot bulb *C*, carrying with it some of the vapor of the oil that has fallen on the projecting lip; and near the end of the compression stroke the pressure and temperature conditions in the vaporizer will cause ignition and explosion. The presence of the steam reduces the explosion pressure and permits a higher compression.

The combustion in an engine of this kind cannot be as complete as in the type where a thorough mixture of the fuel and air can be brought about. Some of the air admitted will remain inactive, as it does not get near the oil. Consequently such engines are comparatively large for the power they develop.

Drawbacks in Use of Vaporizers. While oil engines with internal or external vaporizers are quite simple and low in first cost, their method of vaporization is rather crude and has been found in actual use to be open to objections which are the cause of a common prejudice against such engines. The chief drawback to all these vaporizers is the practical impossibility of vaporizing the fuel completely at all loads and under all conditions and of maintaining the chamber at a temperature which is always sufficiently high to vaporize all of the oil, but, on the other hand, is never hot enough to decompose it. Moreover, the combustion is often incomplete, and the efficiency low. This manifests itself by the objectionable smoke and odor of the exhaust gases. In order to obtain certainty of ignition in engines with internal vaporizers, and at the same time prevent pre-ignitions at different loads, the temperature of the vaporizer should vary with the load, which is found to be a practical impossibility. The pre-heating of the mixture, as required for engines with external vaporizers, decreases the weight of the air aspirated, and therefore the capacity of the engine, while the throttling of the air in passing through the vaporizer chamber and passages, as well as the high back pressure due to the exhaust gases passing through the jacket space of the vaporizers, decreases the power output of such engines still more. The necessity of first heating the vaporizer externally by means of a lamp before the engine can be started is rather inconvenient, as it takes at least from five to ten minutes. The fuel consumption of these engines averages about 1 pound of oil per b.h.p. hour, corresponding to a thermal efficiency of not over 15 per cent and never exceeds 20 per cent, corresponding to a consumption of $\frac{3}{4}$ pound per b.h.p. hour.

ATOMIZERS*

Diesel Methods Give Improved Vaporization. The Diesel-type engine overcomes practically all of these difficulties in the use of

*Abstract of Article by H. R. Setz, *Journal of A.S.M.E.*, October, 1911

liquid fuel, whether heavy or light, and particularly in the case of the heaviest liquid fuels. In this type, fuel is not admitted to the combustion space until the charge, which consists of air only, is compressed to a pressure of about 500 pounds, with a resulting temperature which is sufficient to ignite any liquid fuel injected into it. Fuel is gradually injected, at the end of the compression stroke, by means of a cooled air blast which is at a pressure of 250 to 500 pounds above compression pressure in the cylinder. This high-pressure air blast, with proper form of the atomizer and injection nozzle, completely atomizes the fuel during the injection period and carries its small particles directly into the highly compressed and heated air in the cylinder, where they are at once vaporized and ignited.

The heavier the fuel used, the more finely must it be atomized in injecting it into the compressed charge, in order to insure complete vaporization and ignition. If the heavier oils are injected in large particles, only the surface of the particle will be burned, while the center will be converted by the heat of combustion into a pitchy substance, which will be deposited on the cylinder walls and valves. Since, with a properly designed atomizer, the oil particles are completely burned immediately after their mixture with air, there is no possibility of deposits forming on the cylinder walls, and combustion is so complete that the exhaust is smokeless and without odor.

Efficiency. Numerous tests of different sizes of this type of engine show an average fuel consumption of less than ½ pound of oil per b.h.p. hour, corresponding to a thermal efficiency of about 30 per cent.

Classification of Atomizers. While the oil is in all cases atomized by the action of the injection air in forcing it through the injection nozzle, this result is accomplished in two different kinds of apparatus—the *closed injection nozzle* and the *open injection nozzle*. When the closed injection nozzle is used, special atomizers or distributors are placed in front of it in order to distribute the oil properly and to direct the injection air so as to facilitate complete atomization. In the open injection nozzle no special atomizer is used, the atomization being secured entirely by means of the action of the injection air on the oil in its passage through the nozzle. The closed injection nozzle was the type used on the original Diesel engine, and is the type almost exclusively used in this country.

The open injection nozzle has been developed in Germany during the last five or six years, especially for horizontal engines.

Closed Injection Nozzles. *Busch-Sulzer Atomizer.* The atomizer shown in sectional view in Fig. 39 is arranged horizontally on the side of the combustion chamber. Owing to this horizontal position, particular care must be taken to distribute the oil equally around the circumference of the injection valve, otherwise the fuel oil will flow to the lower part of the annular space adjoining the atomizer and, when the injection valve opens, the injection air will rush into the cylinder through the upper atomizer openings, which are not covered by the oil. This condition obtains particularly at

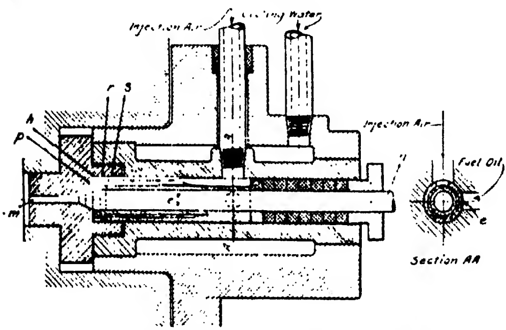


Fig. 39. Section of Atomizer of Busch-Sulzer Brothers Diesel Engine
Courtesy of American Society of Mechanical Engineers

light loads, when the amount of oil sent to the atomizer by the fuel pump is reduced. This condition is unfavorable to good operating results and economy, as the fuel is not properly atomized, and the combustion may be checked by the blast of relatively cold injection air, which is unaccompanied by particles of fuel oil.

In the atomizer of Fig. 39, oil and injection air come together in space *s*, the oil entering along passage *e*, and annular ring space *r*, through a ring of holes *h*. As the injection valve *n* opens, air and oil, divided into small streams by a circle of holes *p*, are forced into the injection nozzle *m*, where these streams impinge upon each other, atomizing the fuel.

Fulton-Tosi Atomizer. The atomizer shown in Fig. 40 is used on one of the newer American Diesel engines, and is also typical of those used on European engines. A series of plates *b*, arranged just below space *s* around the injection valve guide *g*, are provided with small holes in such a way that they straddle each other from plate to plate. These plates help to retain the oil after having been deposited in space *s*, while the holes will distribute it equally and mechanically divide the blast of injection air into small streams, thus disintegrating the fuel passing down through them. By means of passages *p* arranged in the circumference of plug *l*, these streams are directed into the injection nozzle *m*, where they acquire their maximum velocity. The resistance of the oil against the abrupt acceleration thus produced causes the oil to be disintegrated into small particles, which are carried directly into the body of highly heated air in the combustion chamber. This atomizer is generally placed vertically in the center of the cylinder head, and thus, even at low loads, the atomization is perfect.

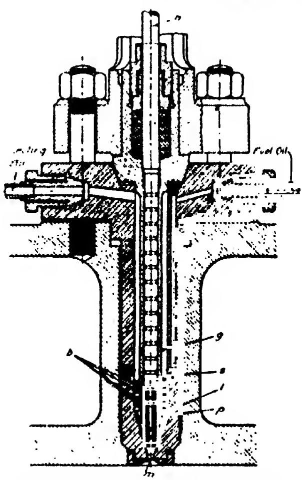


Fig. 40. Atomizer of Fulton-Tosi Diesel Engine
Courtesy of American Society of Mechanical Engineers

De La Vergne Type "FH" Atomizer. Fig. 41 shows constructional details of the injection valve and atomizer used on a horizontal modified Diesel engine. Oil and injection air come together in annular space *s*, formed between the injection valve guide *g* and cage *a*. As the injection valve *n* opens, oil and air proceed along the outside of guide *g*, and are forced to pass through a series of chambers connected by a system of fine diagonal channels *d*, on the

outside of *g*. The oil is thus distributed equally around the circumference of needle-valve guide *g* and enters injection nozzle *m* in a state of fine subdivision, and then is blown into the combustion chamber and hot bulb.

Atomizer Adjustments in Closed Injection Nozzles. In the closed injection nozzle, the oil can be deposited in the atomizer at any part of the cycle, since at all times the oil must be forced in against the pressure of the injection air. In practice, the oil is generally delivered at the start of the compression stroke. Since the fuel and the injection air come into contact with each other before the actual injection period, the atomizer cage, as well as the injection air, must be well cooled in order to prevent pre-ignitions or the formation of deposits due to partial vaporization of the fuel. The

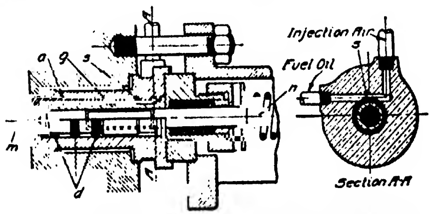


Fig. 41. Type "FII" Atomizer for De La Vergne Oil Engine
Courtesy of American Society of Mechanical Engineers

cooling of the cage may be accomplished either by water-jacketing the cage itself or by locating it in a well-cooled portion of the cylinder or cylinder-head casting.

The points of opening and of closing the injection valve remain unchanged at all loads, i.e., the length of the period the injection valve is open is constant unless the lift can be varied by hand, as is the case in some engines. Within this period a variable quantity of fuel, according to the load, is injected. To accomplish this most satisfactorily, it is usual to increase the pressure of the injection air with increasing loads on the engine, i.e., with increasing amounts of fuel to be injected. Diesel-engine manufacturers recommend a pressure increase of about 250 pounds from light to maximum load. The compression in the engine cylinder is constant at all loads, so that the resistance to injection is constant, but the amount of fuel

TABLE XII

Variation of Injection Air Pressures with Varying Loads (H. R. Setz)

Engine Load, B.H.P.	Injection-Air Pressures, Lb.	Injection-Air Compressor	
		i.h.p.	Per Cent Engine Load
300	950	19.3	6.4
250	865	18.3	7.4
185	830	18.4	10.0
145	790	19.0	12.8

which must be accelerated and atomized by the injection air varies with the load. If the injection-air pressure is too high at light loads, the relatively cool injection air may chill the hot compressed charge before any of the oil is vaporized and ignited, and thus lower the temperature of the charge and endanger the certainty of ignition, sometimes to the extent of a complete "miss", or causing incomplete combustion. To insure certain ignition, oil particles must be injected with the first particles of injection air, and these must be vaporized and ignited before enough cold injection air enters the combustion chamber to cool the air there materially. With very heavy oils, such as coal-tar oils, ignition may be insured by injecting first a very small charge of a lighter or "ignition" oil, immediately followed by a charge of the heavier oil. The variation of injection-air pressures with varying loads on a 4-cylinder 250-horsepower engine is given in Table XII, which also shows the indicated compressor work. No arrangements have so far been made on stationary engines to vary automatically the pressure of the injection air according to load variations; this must be done by hand, at the judgment of the engine operator.

Improved Injection Arrangement on Sabathé Diesel Engine. In a modification of the Diesel engine recently brought out in France and known as the Sabathé motor, there is an attempt to eliminate the inconvenient requirement of variable injection-air pressures with varying loads. Its fundamental features are identical with those of the Diesel engine, with the exception that not only the delivery of oil, but also the lift of the injection valve n, are varied by the gover-

nor according to the load on the engine. Constructional details of the injection valve and nozzle are shown in Fig. 42. In addition to the injection valve *n*, a second valve *v*, sliding on *n* and ordinarily held down on its seat by spring *l*, is provided. This valve *v* is lifted by collar *r* on the injection valve stem when the lift of the latter is sufficient. On light loads only enough oil is delivered by the oil

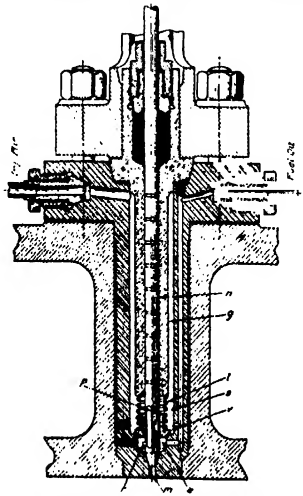


Fig. 42. Atomizer of Sabathé Engine
Courtesy of American Society of Mechanical Engineers

pump to fill the chamber *e* underneath the valve *v*. This is blown into the cylinders when the needle valve *n* lifts, the injection air passing down groove *p* in the needle valve stem. On heavier loads the amount of fuel delivered by the pump fills the chamber *e*, and overflows into space *s*, the oil contained in the chamber *e* being injected first and followed by that contained in space *s*. The pressure of the injection air is maintained at 800 pounds.

Open Injection Nozzle. The modified Diesel, or "open" injection nozzle, was developed to simplify the apparatus. With this

nozzle, oil is delivered to the space *s*, Fig. 43, when the piston begins its compression stroke, i.e., when the pressure in the engine cylinder and in the space *s* is low. The oil has to lie in space *s* throughout the compression stroke in contact with surfaces and air which attain high temperatures. This may result in partial evaporation of the fuel and premature ignition, if the fuel contains components of low volatility, and also in the formation of deposits.

Construction details of a typical injection nozzle and air-admission valve are shown in Fig. 43. No atomizer is used, the oil being blown directly from the space *s* through the injection nozzle *m* into

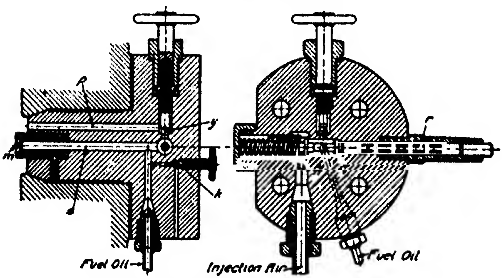
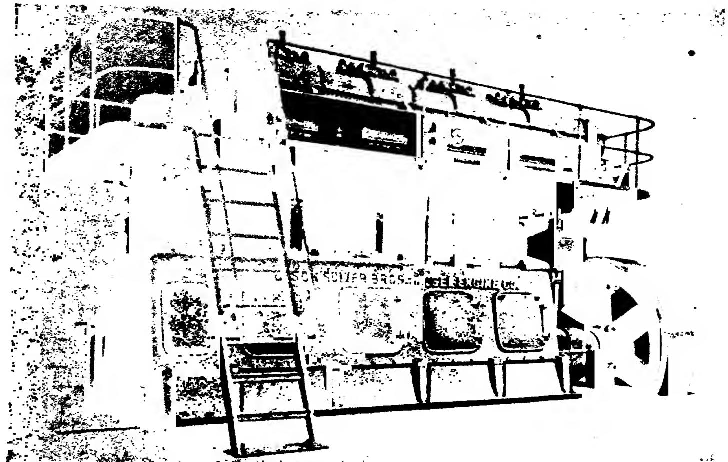


Fig. 43. Typical Open Injection Nozzle and Air Valve
Courtesy of American Society of Mechanical Engineers

the cylinder. The air-admission valve *n* is operated by a push rod *r*, provided with a valve *v* on its inner end, which prevents any leakage along the rod *r*, except during the very short interval when the air-admission valve is open. No stuffing box, such as is used on the injection valves *n* of the original Diesel engine, Figs. 39 and 40, is necessary. At heavy pressures, the valves may be easily prevented from closing properly by excessive tightening of the glands, thus causing loss of injection air and even premature ignitions.

Air of approximately the same pressure as injection air is used for starting Diesel engines. The injection nozzle and the starting device are combined in Fig. 43 in a compact and simple arrangement. By opening the by-pass valve *y*, communication to the cylinder is established through the passage *p*, as well as through the open injection nozzle *m*, and enough air is admitted to start the engine.



TYPE B, FOUR-STROKE CYCLE, FOUR-CYLINDER DIESEL ENGINE
Courtesy of Busch-Sulzer Brothers—Diesel Engine Company, St. Louis, Missouri

MODERN INTERNAL COMBUSTION ENGINES

Classification. With the improvement of design and reliability of internal-combustion engines there has come an astonishing increase in the breadth of the classification of these types. Units of a size hitherto untried have been designed and perfected. The development of the Diesel types has also had its effect particularly in the field of marine engineering. Again the small units for farm use have multiplied enormously, and yet always with a steady increase in efficiency and reliability. Modern internal-combustion engines may be divided broadly into three general classes: the Otto-cycle gas engine; the low-pressure oil engine; and the Diesel or constant-pressure combustion engine.

Otto-Cycle Gas Engine. The Otto-cycle gas engine is further divided into three general classes:

(1) Moderate-Power Stationary Engines. These are gas engines for stationary purposes, of all powers up to about 200 horsepower in a single cylinder. These engines are characterized by longer strokes and moderate speeds, by greater weight, and by the use of a governor. They show an extraordinary variety in form and arrangement, although, like the high-speed engines, they are practically always single-acting. They are also made to use any of the liquid or gaseous fuels. The ignition is usually by electric spark, though rarely of the jump-spark type.

(2) Large Gas Engines. The large gas engine class includes all engines which are capable of developing 250 horsepower and over in a single cylinder. These engines are the latest developments in gas-engine practice. They are horizontal, double-acting, and have water-cooled pistons and rods. They use the low-tension electric ignition system. The fuel most commonly used in them is blast-furnace gas, though producer gas, coke-oven gas, and natural gas are sometimes used.

(3) High-Speed Engines. These are employed principally in automobiles and motorboats, developing generally not more than 15 horsepower in a single cylinder. They are usually vertical and multicylinder, using gasoline as fuel and having jump-spark ignition. This highly specialized type has had an enormous development in the past few years, and has practically reached a standard form and

proportions. It is of extreme lightness and compactness, runs at high speeds, and has no governor.

The term "gas engine", as here used, does not necessarily signify that the fuel used is a gas in its original form, but that it is a gas when introduced into the cylinder, i.e., any of the fuel gases or the vapor from the more volatile liquid fuels—gasoline, naphtha, benzine, or denatured alcohol—which have been vaporized or gasified in an apparatus outside of the engine proper.

Low-Pressure Oil Engine. Low-pressure oil engines may be subdivided in the same manner as gas engines. In this class are included all engines burning a liquid fuel at constant volume which do not fall into the first class. Some of the engines designated as low-pressure oil engines might be regarded as gas engines, since the fuel is treated in an external vaporizer; but the vaporization which takes place outside the cylinder is only partial, the process being completed within the cylinder itself.

OTTO-CYCLE GAS ENGINES

MODIFICATIONS OF OTTO CYCLE

Increasing the Compression. As the efficiency of the Otto cycle has been shown to depend on the amount of compression, the obvious way of increasing the efficiency is to decrease the clearance and thereby raise the compression pressure. The amount of compression that can be used is limited in two ways. The first is that it is not commercially practicable to construct engines which will work properly under very high pressures rapidly imposed by explosion. With an engine compressing the charge to 100 pounds pressure and using a strong explosive mixture, the pressure in the cylinder rises suddenly to about 350 pounds; and at present this is about the practicable limit. If the explosive mixture is weak, the compression may be increased. A compression as high as 200 pounds is sometimes used with very weak mixture and results in a maximum pressure of about 300 pounds.

The second objection to the use of high compression is that the rise in the temperature of the mixture resulting from the compression may easily be sufficient to explode the mixture before the piston has reached the end of its stroke. Such pre-ignition of the charge, tend-

ing to force the piston back, gives rise to a great shock, which is very destructive to the engine, reduces its efficiency, and is to be avoided.

Pre-ignition may occur even when low compression exists if any part of the clearance is not water-jacketed, if there is any metallic projection into the clearance space, or if there is a gas pocket. Unjacketed parts or projections, not being properly cooled, are liable to be raised to a temperature high enough to cause the ignition of the charge. This necessitates water-jacketing of the exhaust valve and of the piston in engines of large size. Gas pockets, especially when of considerable length and small diameter, give trouble because the combustion in them is slow as a result of the character of the mixture they contain. The burned gases with which such a pocket is filled at the beginning of the compression stroke are only slightly mixed with the fresh charge. During compression, the fresh charge is forced into the open end of the pocket; but there is still no satisfactory mixture with the burned gases. When explosion takes place the pocket will contain stratified gases, ranging from a strong mixture at the open end to inert burned gases at the other end. As the combustion progresses from the open end, it encounters a mixture which becomes continuously weaker and slower burning. The result may be that the combustion is still going on in the pocket after the admission or even after the compression of the next cycle has started. The fresh mixture may meet the flame and be ignited, either during the suction stroke, when it causes a back-fire, or during the compression stroke, when it causes a pre-ignition.

Scavenging. Another method of increasing the efficiency is by what is known as "scavenging" the cylinder. In the ordinary Otto cycle the charge compressed consists of a mixture of fresh air and gas with the burned gases remaining in the clearance space from the previous cycle. If these burned gases are expelled from the cylinder by a charge of fresh air before the admission of the explosive charge, the force of the explosion and the efficiency are increased. The clearing out or scavenging of the cylinder with fresh air has been accomplished in several ways. The simplest method is by the use of an exhaust pipe of such length that the gases, exhausting from the cylinder with great velocity, create a vacuum in the cylinder near the end of the exhaust stroke. This vacuum causes the automatic air-admission valve to open; and the consequent rush of air from the

air valve to the exhaust port flushes out the cylinder, especially if the air and exhaust valve are on opposite sides of the clearance space. It is, however, impossible to predict with certainty the occurrence of this vacuum, several manufacturers having tried without success to utilize this accidental phenomenon. Occasional scavenging is obtained in engines governing on the hit-and-miss principle, each idle cycle flushing out the cylinder, with the result that the succeeding explosion is of greater force than the normal explosion. Scavenging has also been accomplished by the injection of a blast of high-pressure scavenging air previous to the opening of the inlet valve.

Compounding. It has been pointed out already that the pressure is high at the end of the expansion in the Otto cycle, and that the efficiency of the cycle can be increased considerably if the gas is expanded more completely. Ordinary steam-engine practice suggests that the more complete expansion can be obtained by compounding; but so far no attempts to make a satisfactory compound gas engine have proved successful. The practical method of obtaining more complete expansion is to take into the cylinder a diminished charge. The two methods of accomplishing this are discussed elsewhere. The only fundamental difference between engines using these two methods is that in one case the governor controls the amount of the opening of the admission valve, and in the other determines the instant at which the admission valve shall close.

Double-Acting. One of the main objections urged against the Otto cycle is that it requires two revolutions of the engine for its completion, so that the expansion or motive stroke comes but once in four strokes. A very irregular driving effort results from this, making large flywheels necessary if the main shaft is to rotate uniformly, or else requiring the use of several engines working on the same shaft. The motive efforts can be made twice as frequently if the cylinder is double-acting, with admissions and explosions occurring on both sides of the piston. Many large engines are now being made double-acting; but the practical troubles in keeping the piston, piston rod, cylinder, and stuffing box cool enough for satisfactory working have prevented the use of double-acting cylinders in engines of small size.

Two-Cycle Engines. *Action of Cycle.* An increased frequency of the expansion or motive stroke can be obtained by a slight modi-

fication of the Otto cycle, which results in the cycle being completed in two strokes, and which is consequently called the two-cycle method. Single-acting engines using the two-cycle method give an impulse every revolution, and thus not only give greater uniformity of speed of rotation to the crankshaft, but also develop 60 to 80 per cent more power than *four-cycle* or Otto-cycle engines of the same

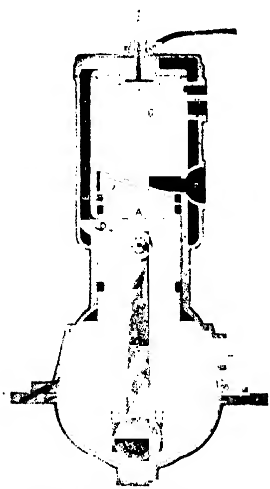


Fig. 44. Smalley Two-Cycle Engine—Piston at Bottom of Stroke

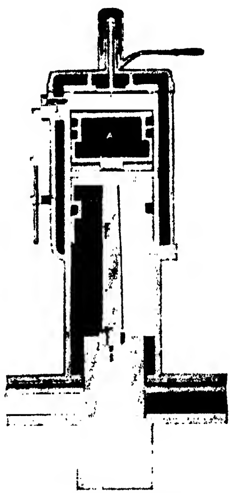


Fig. 45. Smalley Two-Cycle Engine—Piston at Top of Stroke

size. Moreover, they are generally of great simplicity, having fewer valves than the four-cycle engines. An example is shown in Figs. 44 and 45, of a two-cycle engine of small size and of the *two-port* type. Fig. 44 is a vertical section, showing the piston at the bottom of its stroke, and Fig. 45 is a vertical section in a plane at right angles to the previous section plane and shows the piston at the top of its stroke. As the trunk piston *A* makes its upward stroke, it creates a partial vacuum below it in the closed crank chamber *C*, and draws in

the explosive charge through B. On the downward stroke, the charge below the piston is compressed to about 10 pounds pressure in the crank chamber C, the admission through B being controlled by an automatic valve (not shown) which closes when the pressure in C exceeds the atmospheric pressure. When the piston reaches the lower end of its stroke, it uncovers exhaust port K, and at the same time brings admission port D, in the piston opposite the by-pass opening $E E E$, and permits the compressed charge to enter the cylinder G, through the automatic admission valve f as soon as the pressure in the cylinder falls below that of the compressed charge. The return of the piston shuts off the admission through E, and the exhaust through K, and compresses the charge into the clearance space. The charge is then exploded, Fig 45, and the piston makes its down, or motive, stroke. Near the end of the down stroke, after the opening of the exhaust port K, the admission of the charge at the top of the cylinder sweeps the burned gases out, the complete escape being facilitated by the oblique form, Fig. 44, of the top of the piston. The engine is so designed that the piston on its return stroke covers the exhaust port K just in time to prevent the escape of any of the entering charge. The processes described above and below the piston are simultaneous, the upstroke being accompanied by the admission below the piston and compression above it, while the down stroke has expansion above the piston and a slight compression below it. The very short interval of time between the beginning of the exhaust and the admission of the new charge—which enters as soon as the pressure in the cylinder has fallen enough to permit the admission valve to open—makes premature ignition of the charge, or "back-firing", of not infrequent occurrence. If the mixture is weak, or the speed is very high, so that the charge is still burning when admission begins, or if the frequency of the explosions brings any part of the cylinder to a red heat, the charge will be ignited on entering, and the explosion will then travel back through $E E E$ to the crankcase, which has to be made strong enough to resist it. In large engines the charge is compressed by a separate pump, and not in the crankcase.

Single-Valve Type. A modification of the two-cycle engine makes the construction even more simple, so that the only valve on the engine is the automatic valve admitting the charge to the crank-

case. In this engine, Fig. 46, the series of operations is precisely similar to that just described. The only difference is in the by-pass connection E, which has no valve between it and the cylinder. The exhaust is made to open a little earlier than the admission, in order to make sure that the pressure in the cylinder shall have fallen below the pressure of the slightly compressed charge when the admission port opens. If the opening of the exhaust and admission ports were simultaneous, as in the engine just described, some of the exhaust gases would force their way through E to the crankcase, igniting the charge there. The piston is so shaped that the entering charge is directed to the top of the cylinder, forcing out the burned gases before any of the charge can escape through the exhaust port.

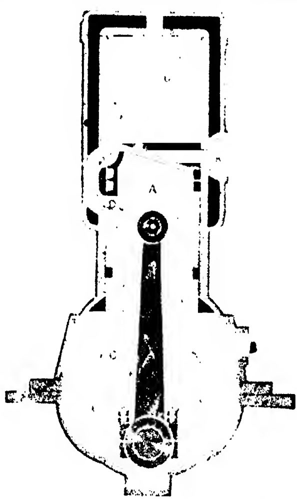


Fig. 46. Modification of Smalley Two-Cycle Engine with Its Valve between By-Pass and Cylinder

Single Revolving-Disk Valve Type. In place of the automatic inlet valve at B, a revolving-disk valve is sometimes used, which turns with the crank and contains a slot that registers with the crankcase inlet during part or all of the upstroke of the piston. The disk is pressed against its seat by a light spring. This arrangement makes the admission of the charge to the crankcase positive and permits of adjustment of the duration of admission, and consequently of the volume admitted. It sacrifices, however, the reversibility of the engine.

Valveless Type. A further and last modification of this engine makes it entirely valveless and of the utmost simplicity. It is illus-

trated in Fig. 47. The admission of the charge is through the port *B*, which is covered and uncovered by the piston, and which consequently does not require any automatic valve. During the upstroke of the piston, a vacuum is created in the closed crankcase, till near the top of its stroke, when the admission port *B* is uncovered and the explosive charge rushes into the crankcase, filling it until the pressure there is approximately atmospheric pressure. The other operations are exactly as in the engine previously described, the charge being compressed in the crank casing during the down stroke, and then transferred through port *D* in the hollow piston, and through

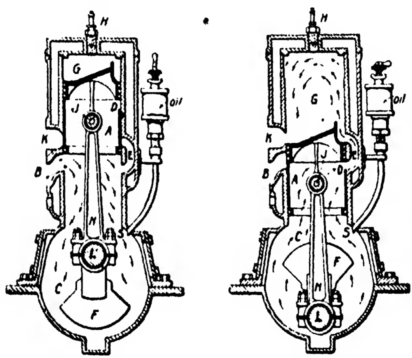


Fig. 47. Smalley Three-Port Two-Cycle Engine

port *E* in the cylinder wall, to the upper side of the piston, when it is near the end of its down stroke. This modification is generally known as the "three-port type of the two-cycle motor".

The power of a small two-cycle engine can be varied by throttling—that is, by varying the amount of the charge taken into the cylinder. This is accomplished either by throttling the admission to the crankcase, or by throttling in the by-pass between the crankcase and the cylinder. There is probably little to choose between these two methods.

Loss of Efficiency in Two-Cycle Types. The great compactness and simplicity of the small two-cycle motor are obtained at some cost

of efficiency. In all gas engines a certain amount of work has to be done in getting the explosive mixture into the cylinder during the suction stroke, and in expelling the exhaust gases during the exhaust stroke. This gas-friction work is represented on the indicator card of an Otto-cycle engine by the negative loop, Figs. 3 and 8, which has to be subtracted from the positive loop in order to give the indicated horsepower of the engine. In the four-cycle engine, this negative work is usually from 2 to 5 per cent of the total work, and is a dead loss. In the two-cycle engine, considerably more work must be done in order to get the gas into the cylinder. The time available for the admission of the charge is extremely short. In a small high-speed engine, it will be from one- to two-hundredths of a second; in a large two-cycle engine, it may amount to one-twentieth of a second. In any case, it will not be more than one-third to one-fifth of the time available for admission in a four-cycle engine. Moreover, this admission takes place while the exhaust gases are going out rapidly and, consequently, while the pressure in the cylinder is appreciably greater than atmospheric pressure. In order to overcome the back pressure of the exhaust, and also in order to be able to enter with the very high velocity necessitated by the short duration of admission, the explosive mixture has to be pre-compressed to 8 or 10 pounds above atmospheric pressure before its admission to the cylinder. Whether this pre-compression is done in the crankcase, as in small engines, or in separate compression pumps, as in large engines, it requires the expenditure of a considerable amount of work—an expenditure which decreases the available power of the engine without giving anything in return, other than the possibility of maintaining the cycle of operations. This loss of power in compressing the charge is ordinarily from 7 to 12 per cent of the total work done in the cylinder.

Another loss of efficiency in the two-cycle engine results from the fact that the admission and exhaust ports are open at the same time. An endeavor is made to have the exhaust port close before any of the entering charge has reached it; but practically it is not possible to accomplish that particularly in an engine which is to run at various speeds. If, in an endeavor to prevent such loss of gas direct to the exhaust, the exhaust port closes early, too large a volume of the exhaust gases will be retained in the cylinder; the amount of the

charge which can enter will be correspondingly decreased; and both the efficiency and the capacity of the engine will suffer. In larger engines this trouble is obviated to a great extent by forcing air into the cylinder slightly ahead of the explosive charge and closing the exhaust port when the charge of fresh air is passing through. This device is also valuable in preventing back-firing of the charge.

Besides its simplicity and compactness, the small two-cycle engine may claim reversibility as one of its advantages. The direction of rotation in the small valveless two-cycle engine is determined solely by the timing of the ignition. It is possible to reverse such a motor merely by making the point of ignition very early. This causes an explosion well before the ending of the compression stroke,

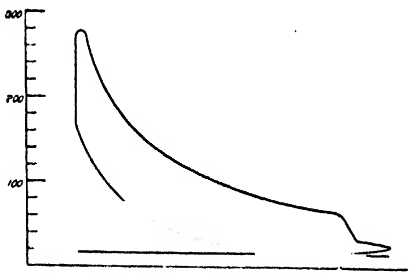


Fig. 48 Indicator Card of Two-Cycle Engine

and may develop sufficient pressure to stop the piston before it gets to the end of the stroke and start it going in the other direction. When once started in the other direction, the ignition, if unchanged, will be a very late ignition, giving comparatively small power; shifting the ignition back a little will give the engine its full power in its reversed direction. This process is practicable only in small engines with light reciprocating parts, and is most convenient for small motor-boat use.

The two-cycle engine develops on the average about 70 per cent more power than a four-cycle engine of the same size and speed; it uses from 10 to 20 per cent more gas per brake horsepower. A typical indicator card for a two-cycle engine is shown in Fig. 48.

MODERATE-POWER STATIONARY ENGINES

Vertical Type

As the moderate-power stationary engines are of most general importance to the engineer, the descriptions of engines given immediately below are taken from that class. The greater part of what

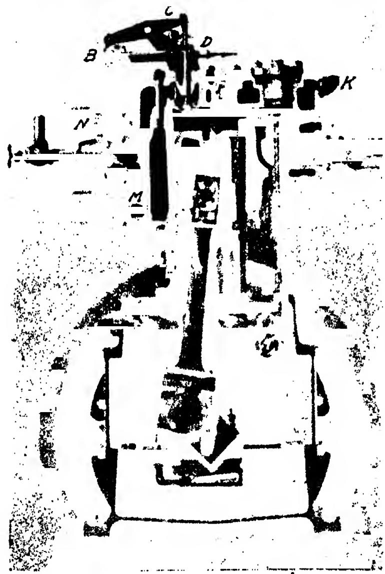


Fig. 49. Westinghouse Vertical Gas Engine, Medium Size
Courtesy of Westinghouse Machine Company, East Pittsburgh, Pennsylvania

follows applies to gas engines of every kind; where it does not, special attention is called to that fact.

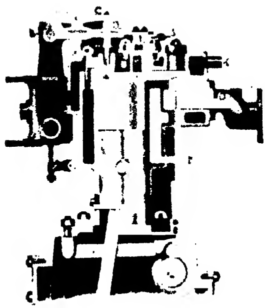
Westinghouse Single-Acting Engine. The construction of a medium-sized gas engine using the Otto cycle is illustrated in the sectional elevation, Fig. 49, of a vertical engine. As in practically

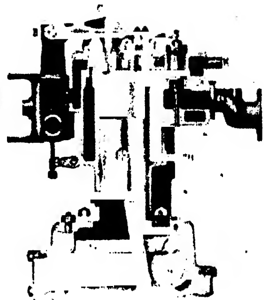
all such engines, the engine is single-acting and has a long trunk piston, which acts as a crosshead and also permits the use of several piston rings whereby leakage past the piston is prevented even with the high pressure obtained by the explosion. The engine is made single-acting because the piston, piston rod, and stuffing box give great trouble if exposed to the high temperature of the burning gases unless they are water-cooled, and the water-cooling of these parts is difficult in small engines. Since the cycle occupies two revolutions, the valves and igniter have to operate once in two revolutions; therefore, the cams which drive these parts are mounted on shafts running at one-half speed of the main shaft.

General Details. Referring to Fig. 49, *A* is the shaft which carries the exhaust-valve cam, and is driven by gears from the main shaft. The exhaust cam works against a roller carried on the free end of the guide lever *G*. The exhaust valve *E* has a long stem projecting downward and resting on a hardened steel plate on the upper side of the guide lever *G*. The spring surrounding the stem serves to bring the exhaust valve back to its seat, and to keep the stem in contact with the guide lever. From the exhaust camshaft *A*, a horizontal shaft with bevel gears leads to the opposite side of the engine, engaging with a vertical shaft, which in turn drives the upper camshaft *B*. The governor is mounted on the vertical shaft. The upper camshaft carries two cams, one of which engages against a roller on the end of the horizontal lever *C*. As the throw side of this cam comes uppermost, the opposite end of the lever *C* depresses the stem of the inlet valve *J*, opening the latter for the admission of the mixture of gas and air. A spring on the stem of the inlet valve furnishes a means for closing it and keeping the cam and roller always in contact with each other. Immediately adjacent to the inlet-valve cam is the igniter cam, which, at the proper instant, operates a horizontal plunger working through the guide *D*, and breaks the electric circuit at the terminals of the igniter *F*.

Water-Jacketing. The cylinder heads and the upper end of the cylinder are thoroughly water-jacketed, as, owing to the high temperatures to which these parts are subjected, they would soon become red-hot if no means were provided for keeping the temperature down. The cooling water enters at *H*, and is discharged at *K*.

Fuel Mixing. The gas and air enter the mixing chamber *M* by





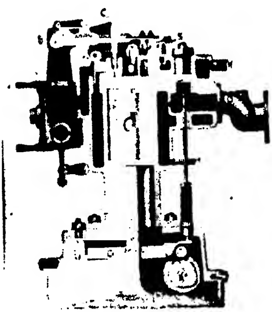


Fig. 52. Gas Engine Valve Set for Ignition

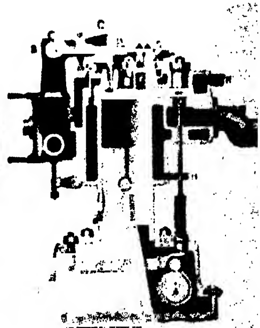


Fig. 53. Gas Engine Valves Set for Exhaust

separate inlets in proportionate amounts which can be regulated, and the mixture is conducted through a distributing chamber to the port *N* leading to the cylinder head in which the inlet valve is located. The exhaust gases escape through *O*.

Operation. The operation of this engine is illustrated in the accompanying figures. The admission of the charge of air and gas takes place during the first downward stroke of the engine, Fig. 50. The exhaust valve *E* is closed, and the admission valve *J* is open, closing only when the piston is at the end of the stroke and the cylinder is full of the explosive mixture.

During the return stroke, Fig. 51, both valves are closed, and the charge is compressed till at the end of the stroke it occupies only the clearance space. Shortly before the end of the stroke, the igniter cam has brought the igniter terminals into contact, completing an electric circuit. When the crank is nearly on its dead center, the igniter terminals are separated by the action of a coiled spring in the guide *D*; and, as they fly apart, the circuit is broken and a spark passes between the terminals, Fig. 52, igniting the charge. An immediate rise of pressure occurs, and the piston is forced downward, both valves remaining closed until just before the end of the down stroke, when the exhaust valve *E* opens.

During the whole of the last return stroke, Fig. 53, the exhaust valve *E* remains open, and the products of combustion are forced through *O* to the atmosphere. The exhaust valve closes as the piston completes the stroke, and everything is ready to repeat the cycle.

Nash Engine. Another form of vertical engine in which the inlet and exhaust valves are located side by side is shown in vertical section in Fig. 54. The inlet valve *a* and the exhaust valve (not shown) are operated from the shaft *c*, which is driven from the main shaft by spur gearing at one-half the speed of the main shaft. A cam on the shaft *c*, acting through a roller, lifts the pivoted lever *d*, at the end of which is the long spindle of the valve *a* through which the charge is admitted. The exhaust valve is behind the inlet valve, and is operated in the same manner. The air and gas are mixed, and the amount of the mixture is regulated in the balanced-disk throttling governor valve *e*, the position of which is regulated by the governor *f*. The air and gas are admitted to the governor valve through the pipes *g* and *h* (not shown), and the proportions of the mixture are

regulated by the hand valves *i*. The exhaust passes out through the water-cooled header *j*. The igniter is a magnetically operated make-and-break plug, timed by a commutator mounted on the camshaft.

Rathbun Engine. The engine shown in Fig. 55 has both the inlet and exhaust valve in the cylinder head, thus permitting the use of a dished cylinder head. The valves are actuated by a device known

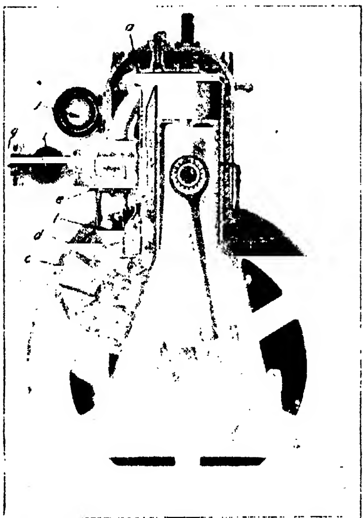


Fig. 54. Vertical Section of Nash Gas Engine
Courtesy of National Meter Company, New York City

as a "roller-path lever". In this device a lever is pinned at one end to the valve-actuating rod, and on the other is fitted with a roller which rests on the top of the valve stem. The top of the lever is curved. Above it is a block which is firmly held in position in the valve bonnet and has somewhat smaller curvature than that on the lever. This block is adjustable in order to allow for taking up wear

in the valve mechanism. When the valve-actuating rod is pushed up to open the valve, the curved surface on the lever rolls smoothly

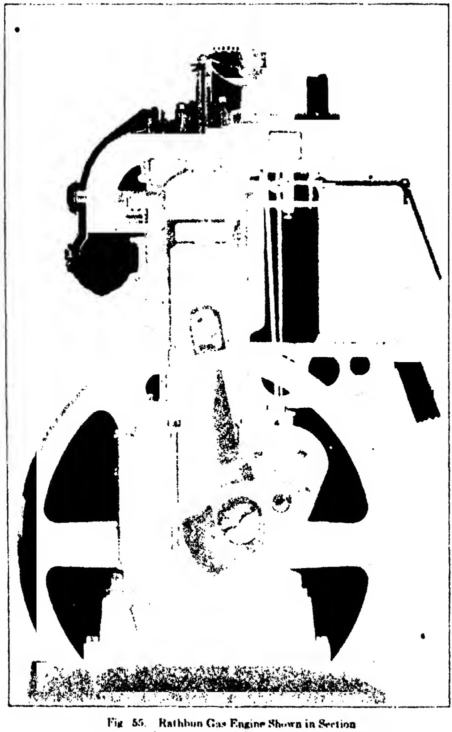


Fig. 55. Rathbun Gas Engine Shown in Section
Courtesy of Rathbun-Jones Engineering Company, Toledo, Ohio

and noiselessly over the curved surface of the block and depresses the roller end of the lever, thereby opening the valve. The actuating rod is connected to an eccentric and operates the rod through a cross-

head which slides in a guide held in the top of the frame or housing. The exhaust valves, exhaust elbows, and manifold are water-cooled. The piston is provided with an air-insulating space or pocket at its top to prevent the accumulation and carbonization of oil on the hot piston head. The journal boxes rest on wedges which are adjustable from the outside by a bolt on either side of the engine, which

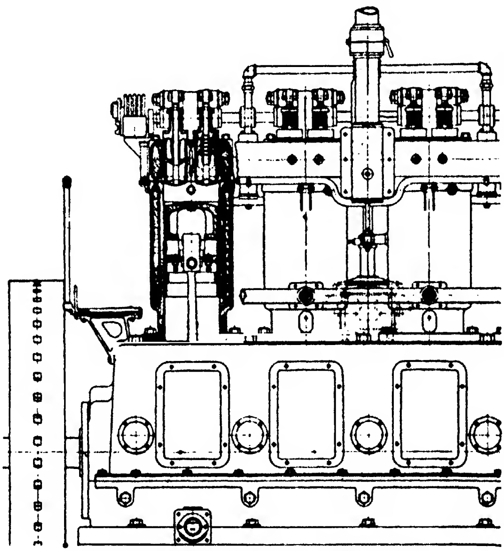


Fig. 56 Front Elevation of Bruce-MacBeth 4-Cylinder Gas Engine
Courtesy of Bruce-MacBeth Engine Company, Cleveland, Ohio

thus facilitates the lining up of the crankshaft. These wedges may be taken out sideways into the engine base by removing the adjusting wedge bolts. This allows the journal box to be removed in a like manner, while the cap may be removed in the usual manner; thus providing a means of replacing the main journals without disturbing the crankshaft. The journal-cap studs are continued upward

through the housing, forming stay bolts, through which all strains created by the pressure in the cylinders are transmitted directly to the main journals. Because of this construction, it is possible to have large doors in the housing to facilitate repairs and adjustments inside

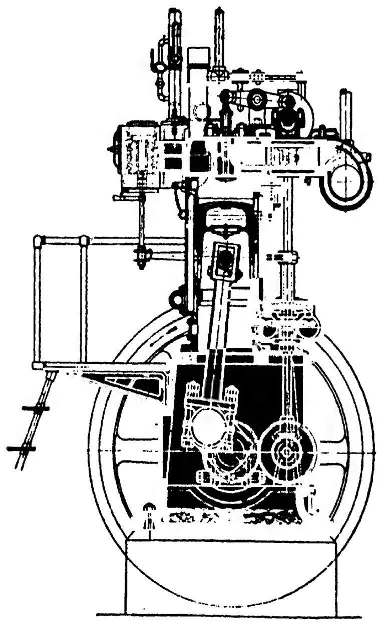


Fig. 57. Side Elevation of Bruce-MacBeth Gas Engine
Courtesy of Bruce-MacBeth Engine Company, Cleveland, Ohio

the crankcase. The governor, of the fly-ball type, is driven from the eccentric shaft and acts directly on a throttle valve. The governor also changes the timing of the ignition to suit the compression as affected by the throttling. The throttle valve is a two-disk balanced valve acting directly on the mixture. The quality of the mixture is

regulated by an additional valve in the gas line, which has light-load and full-load adjustments similar to some carbureters, thus providing an automatic maintenance of the proper mixture throughout the range of load. The ignition is by mechanical make-and-break igniters.

Bruce-MacBeth Engine. The engine shown in Figs. 56 and 57 also has both the inlet and exhaust valve located in the cylinder head The valves are actuated from a camshaft through levers pivoted on brackets bolted to the cylinder heads. The camshaft is carried at the level of the top of the cylinder head in bearings which are an integral part of the jacketed exhaust outlets. The camshaft is driven from the crankshaft through a vertical shaft by bevel gears. The valves are carried in cages, the exhaust-valve cage being water-cooled though the exhaust valve itself is uncooled. The inlet-valve cage is divided into two compartments by a ledge at the middle of the cage—the upper compartment connecting to the gas main and the lower to the air main. A disk is carried on the inlet-valve stem which seats on the dividing ledge of the cage when the inlet valve is closed, thus serving to shut off the gas compartment from the air compartment and preventing the formation of an explosive mixture until the inlet valve is opened. This arrangement helps to scavenge the cylinder since, when the inlet valve opens, air is nearest to the opening, and consequently a layer of pure air is drawn into the cylinder, previous to the commencement of the suction stroke, by the inertia of the exhaust gases in flowing out of the exhaust valve. This layer of pure air tends to replace the products of combustion remaining in the clearance space and to force them out through the exhaust valve, thereby scavenging the cylinder. The exhaust valve closes before any of the oncoming explosive mixture can be lost into the exhaust.

The regulation is obtained by a fly-ball governor mounted on the vertical shaft, which operates either a balanced-cage throttle valve or a balanced-disk throttle valve, depending upon the fuel used. The ignition is obtained by two sets of jump-spark plugs mounted one on either side of the cylinder. One of these plugs is a standard grounded plug and the other is a special plug with both electrodes insulated. The current is obtained from two high-tension magnetos—one for each system—mounted one on either end of the camshaft. A timer or distributor, to direct the current to the proper plug at the proper instant, is built into each magneto.

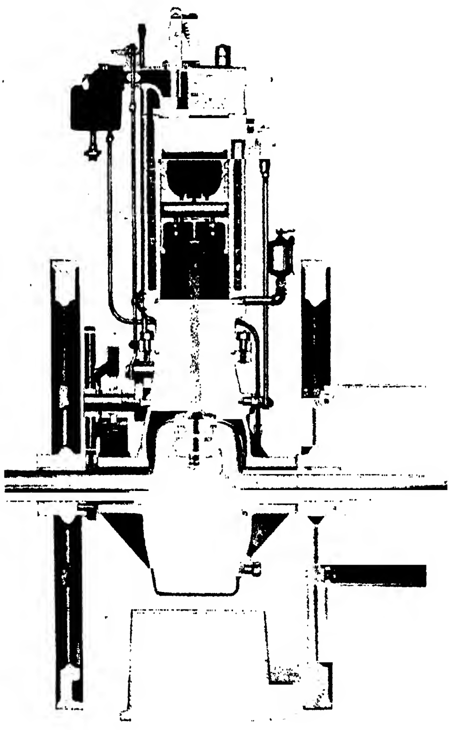


Fig. 58 Fairbanks-Morse Type "T" Vertical Gas Engine
Courtesy of Fairbanks, Morse and Company, Chicago

Fairbanks-Morse Engine. The engine shown in Fig. 58 has both valves located in the cylinder head, the exhaust valve being mechanically operated, while the inlet valve is automatically operated by the suction of the engine. The engine, as represented, is equipped to operate on gasoline.

Horizontal Type

Otto Engines. An example of the horizontal form of gas engine is given in Figs. 59 and 60, which are vertical cross sections through the cylinder and valves. The admission and exhaust valves *A* and *I*

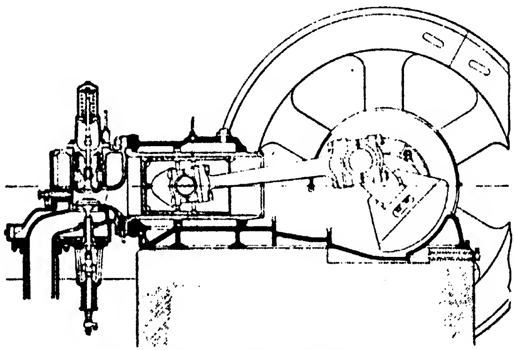


Fig. 59. Otto Gas Engine, Longitudinal Section of Horizontal Type
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

are placed one above the other—the exhaust below. The valves are opened by cams *M* and *H*, respectively, mounted on a horizontal side shaft parallel to the axis of the cylinder. The exhaust cam acts on the end of a lever *L* with a fixed fulcrum, giving an invariable opening to the exhaust valve. The admission cam acts on the end of a lever, *G*. The governor controls the time and duration of the opening of the admission valve and thereby controls the amount of the explosive charge admitted to the cylinder. The ignition is obtained by an oscillating magneto and make-and-break igniter plug, which are operated by a rod driven by a crank on the end of the side shaft. A gas valve *B* is mounted on the inlet-valve stem, as in the previous

engine, only, instead of being mounted rigidly, it is mounted loosely and is held to its seat by a spring.

Alberger Engine. The engine shown in Fig. 61 is a horizontal single-acting tandem. The cylinder head of the front cylinder is provided with a stuffing box to prevent explosions from blowing out

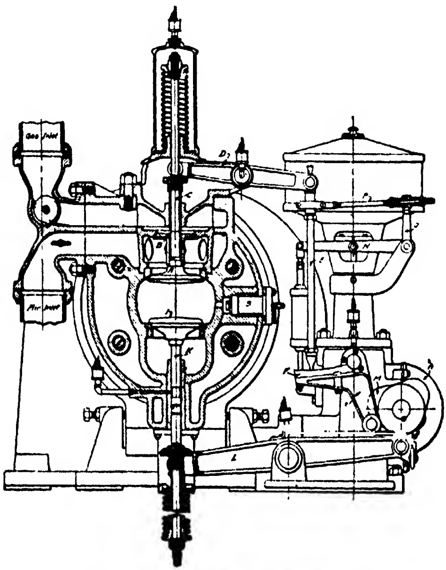


Fig. 60. Transverse Section of Otto Gas Engine
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

around the piston rod. The stuffing box is water-cooled and provided with five snap packing rings. The piston rod is solid and uncooled. The valves are located on the side of the cylinder in a valve chamber, which is a water-cooled casting separate from the cylinder casting. The valves are actuated by a cam and lever as shown in Figs. 158 and 159. The regulation is secured by a Rites inertia flywheel governor

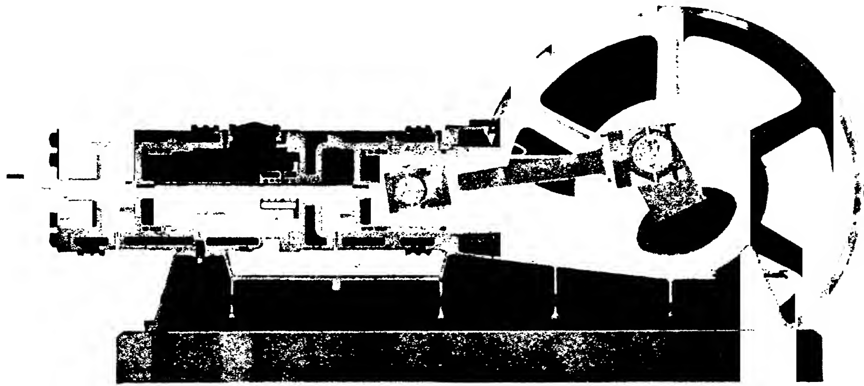


Fig. 61. Alberger Tandem Gas Engine—Longitudinal Section
Courtesy of Alberger Gas Engine Company, Buffalo, New York

rotating a cut-off cage valve. The ignition is obtained by a double high-tension jump-spark system, two spark plugs being provided for each combustion chamber.

Foos Engine. An entirely different arrangement is shown in Fig. 62, where the valves are located on opposite sides of the cylinder *C*. Both are mechanically operated through cams and push rods operating the bell-crank lever shown in the figure. The inlet valve *E* is opened by the lever *A*; the exhaust valve *D*, through the lever *K*. The igniter is placed close to the inlet valve. The provision of a plug above each valve permits ready access to the valves for removal or grinding.

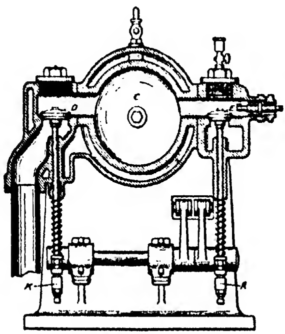


Fig. 62. Foos Gas Engine

LARGE GAS ENGINES

General Characteristics of the Type. A very rapid development has taken place in the size to which gas engines are built, until now there are in operation units developing over 5000 horsepower each. There are some special problems which arise when large power is to be developed in a single engine; and for that reason the large gas engine is here considered separately.

The gases which are commercially available for use in large gas engines are: natural gas, blast-furnace gas, producer gas, and coke-oven gas. Gases rich in hydrogen are objectionable on account of the excessive stresses caused by pre-ignition in a large cylinder.

The American practice is to make large gas engines of the *side-crank* type. With the four-cycle type of 1000 to 1500 horsepower, a tandem double-acting engine is usual; for 2000 horsepower and over, double-tandem double-acting engines are used. The tandem double-acting engine gives two explosions per revolution, resulting in a very uniform speed of rotation.

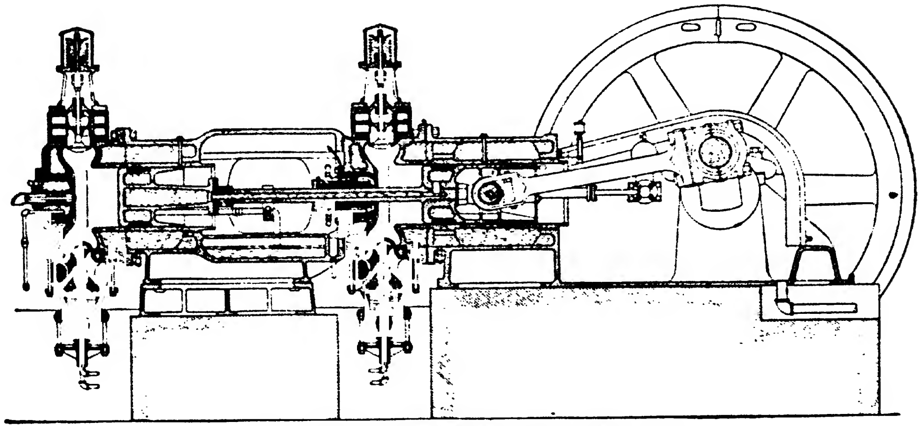


Fig. 63. Longitudinal Section of Warren Heavy-Duty Tandem Gas Engine
Courtesy of Struthers-Wells Company, Warren, Pennsylvania

Warren Engine. An example of a moderate-sized single-acting tandem engine is given in longitudinal section in Fig. 63, and in

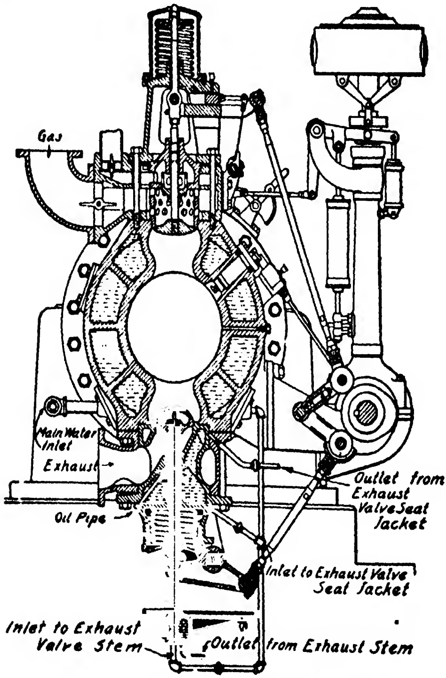


Fig. 64. Vertical Cross Section Through Valves of Warren Tandem Gas Engine, Shown in Longitudinal Section in Fig. 63

cross section through the valves in Fig. 64. This particular engine is no longer built, but many of them are still in service. Besides, this engine is typical of large single-acting tandem engines. This arrange-

ment requires a stuffing box in the front cylinder. The piston and piston rod, the exhaust valves, and the exhaust-valve cages are water-jacketed. The water enters the front piston through a pipe sliding through a stuffing box; goes through the hollow piston rod to the rear piston, and from there discharges through a pipe into a trough in the intermediate bed. The jackets are cast separate from the cylinders, and make sliding joints at their front ends so as to allow of differential expansion of the cylinders and jackets. The jacketing of the exhaust valves is shown clearly in Fig. 64.

The valves are situated vertically above one another, the exhaust, as usual, being below. They are worked by a single cam on the lay shaft, through intermediate push rods and levers; the exhaust valve, through a massive lever with a fixed fulcrum; and the admission valve, through a lever with a movable fulcrum. The position of the movable fulcrum is controlled by the governor through a flexible steel strip; the farther in it is pushed, the less does the admission valve open and, consequently, the more is the charge throttled. The movable fulcrum is quite free to move at all times, except when the admission valve is actually being operated. The governor is free to move even at that time, as it can bend the flexible strip; consequently, the work on the governor is extremely slight, and it can be made very sensitive. These engines have converted 32 per cent of the total heat of the gas supplied to them into indicated work.

Allis-Chalmers Engine. The normal type of large-size four-cycle gas engine is shown in Fig. 65, which is a drawing of the largest gas engine yet built in this country operating on blast-furnace gas, and in Fig. 66, which is a cross section of another engine of the same type. It is a tandem double-acting, side-crank engine, with the inlet and exhaust valves placed as in the engine just described. Owing to the great weight of the pistons and rods, the latter are supported on three slides, and the cylinders are thereby relieved of wear.

Water-Jacketing. The main bearings of the crankshaft are water-cooled, and are provided with spherical seats to allow for deflection of the shaft. Each cylinder with its jacket is a single casting; the great depth of the water-jacket space is intended to permit of differential expansion of the cylinder and jacket; the longitudinal tensile stresses are carried by the jacket wall, the cylinder wall being subjected only to the pressure of the gases. The pistons are

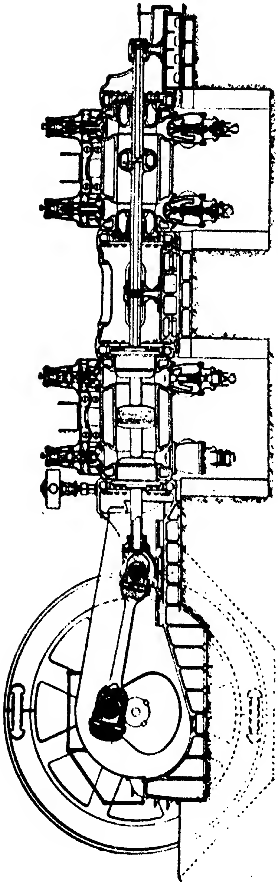


Fig. 65. Longitudinal Section through Allis-Chalmers Double-Acting Tandem Gas Engine

water-cooled; and, as there are no nuts or other projections from the pistons, all parts are adequately cooled. The hollow piston rods are of nickel steel, and are given such an initial camber that they become straight when they support the weight of the pistons and cooling water. Water for cooling the pistons and rods is brought to the central crosshead by means of pipes with swing joints on each side of the crosshead, the water from one pipe going forward, and that from the other pipe to the rear rod. After circulating through the piston and rod, the water is discharged from pipes at the ends of the piston rods into a slot in the tail-guide frame at the one end and, as in the preceding engine, through a pipe which slides through a stuffing box in the main frame. The packing used in these stuffing boxes consists of sectional cast-iron rings enclosed by retaining rings.

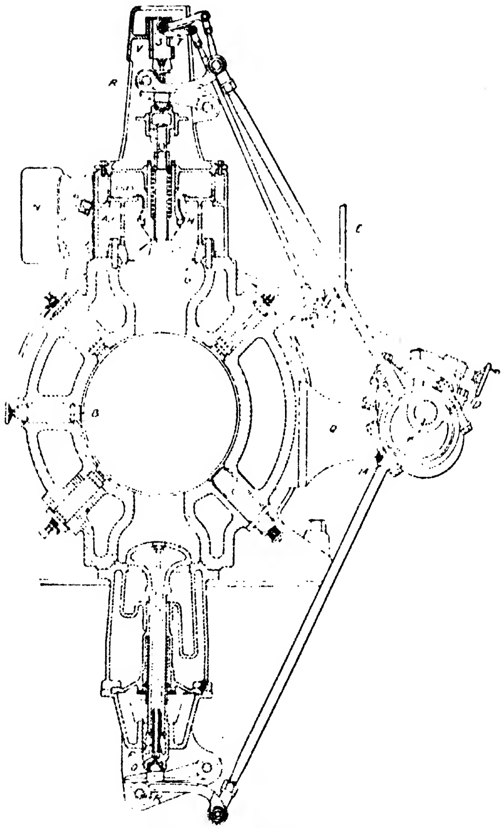


Fig. 66. Vertical Cross-Section through Valves of Allis-Chalmers Double-Acting Tandem Gas Engine

Ignition. Separate gas and air pipes above the engine-room floor are dispensed with by making them an integral part of the cylinder casting at the center of the length of the cylinder, as is shown in Fig. 65. This type of construction is known as a "belted cylinder" and greatly simplifies the appearance of the engine. Because of the large diameter of the cylinders and the consequent large volume of lean blast-furnace gas in the combustion chamber at the instant of ignition, three igniters are used for each cylinder end instead of two, as is the usual practice. Two of the igniters are located, one on either side of the cylinder, forty-five degrees down from the vertical. The third is located in the same plane as the other two on the side of the cylinder away from the lay shaft, forty-five degrees up from the vertical. This third igniter is necessary because of the distance through which the flame has to travel in this engine. Its use increases the power of the engine 20 per cent. The igniters are mechanical make-and-break (shown in Fig. 120 and described on page 183) operated by igniter camshafts, which run at half the speed of the crankshaft.

Valves. In the cross-sectional view through the valves, shown in Fig. 66, some of the details of the cylinder, such as the separate casting for the gas and air passages, are different from those of the engine shown in Fig. 65. The inlet and exhaust valves and the mechanism actuating them are unchanged.

The exhaust valve and its stem are in one piece. The valve stem, head, and cage are all water-cooled, as shown in Fig. 66. The water leaves by a swinging hose connection at *P*, and enters by the internal tube and another hose connection at *O*. The valve is operated by the eccentric *M* on the lay shaft *K*, through the rolling lever fulcrumed at *R*. The rolling lever is the most largely used of the valve-operating devices. The eccentric rod is in tension—an advantage over the push rod, especially in large engines where the total pressure to be exerted on the exhaust valve, in order to start lifting it, is very great.

The inlet valve *G* is operated through the eccentric *L* by means of rolling levers. The period and amount of the opening of the gas valve *H* are controlled by the governor. The quantity of the mixture admitted, and therefore the compression pressure, is kept constant; but the quality of the mixture is varied. The inlet valve *G* opens

before the gas valve *H*, thus admitting air only at first; and, as the exhaust is still open at that time, the entering air scavenges the cylinder and puts out any flame still remaining in it. Then follows the admission of a mixture of air and gas. The gas valve then closes slightly ahead of the main valve, so that the last part of the charge admitted is air; this fills the valve chamber below the inlet valve, but is not in contact with the igniters.

The gas valve *H* is of the double-seated type, and is operated by two rods, shown in Fig. 66, which connect it with the crosshead *V*, and with a rolling lever fulcrumed at *S* on the crosshead *V*. The fulcrum lever *T* is forked on its inner end, and the ends of this fork are pivoted on fixed pins. The rolling lever being connected to the main inlet-valve rolling lever, both levers move in unison. The position of the fulcrum lever *T* is controlled by the governor through the rod *E* and the eccentric on the shaft *F*; consequently, the governor controls the movement of the inner end of the rolling lever fulcrumed at *S*, and of the gas valve *H*.

Starting. Starting is by means of compressed air admitted in turn to each cylinder end, at what would be the expansion stroke, through the valve *C*.

Valve Gear and Governing of Westinghouse Tandem Engine. The valve gear and governing of another type of large gas engine are illustrated in Fig. 67. One eccentric is employed for both the exhaust and the inlet valve. There is a single mechanism which combines the functions of inlet, mixing, and governing valves. The mechanism consists of a spring-operated poppet valve, a stationary cage in which it is housed, and a mixing or regulating valve sleeve which reciprocates within the cage and rotates on the inlet-valve stem. The regulating sleeve is provided with ports registering with corresponding ports in the surrounding valve cage. The valve has three distinct motions—a definite vertical motion of the poppet valve, the vertical motion of the sleeve, and a rotation of the sleeve controlled by the governor alone.

The main poppet valve is worked from the eccentric through a push rod and rolling levers. When it is closed, the cylindrical sleeve closes the air and gas ports. As the main valve opens, the sleeve falls, uncovering the ports. The area of the ports uncovered is determined by the angular position of the sleeve. The rotating sleeve

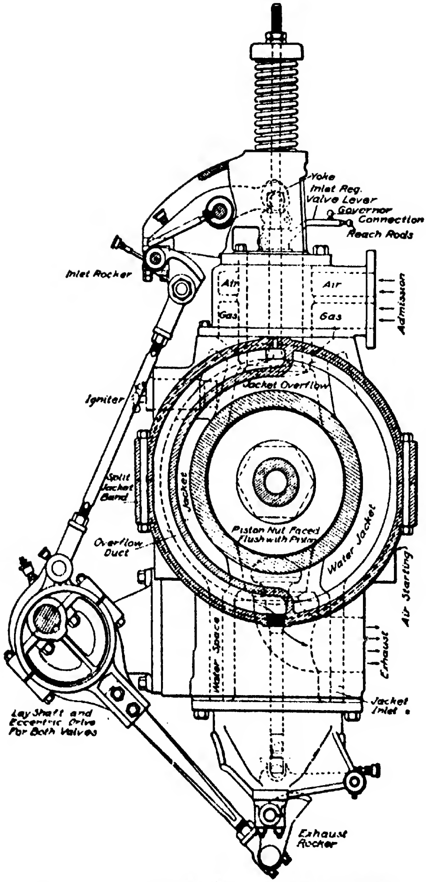


Fig. 67. Vertical Cross Section of Westinghouse Tandem Gas Engine.

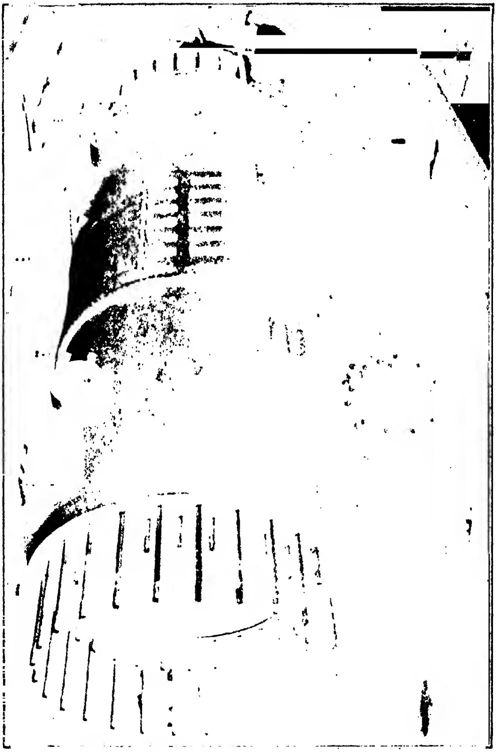


Fig. 68. Cylinder of Large Gas Engine
Courtesy of Westinghouse Machine Company, Pittsburgh, Pennsylvania

acts as a throttling valve, but does not affect the relative amounts of air and gas admitted. The governing is by quantity, not by quality.

General Cylinder Construction. The cylinders of the largest engines of this type are cast in halves fitted together with ground joints. An opening is left between the two halves of the jacket at the center; this opening is closed by a split jacket band, Fig. 68, making water-tight joints with the castings but permitting independent expansion of the cylinder and the jacket.

Effect of Two or More Igniters. It is necessary, in a large gas engine, to have two or more igniters for each combustion chamber. The combustion starts at the igniters, and spreads with a moderate velocity. In order to make it complete in a short time, it is best to start it simultaneously at two or more points. In small engines this is not necessary, as the distance the flame has to go in order to fill the combustion space is short. The effect of using two igniters is shown in Fig. 69, which gives two superposed indicator cards taken from a large engine.

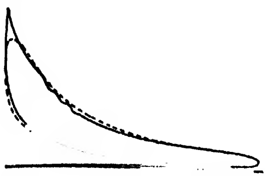


Fig. 69. Indicator Card, Showing Effect of Using Two Igniters

The dotted-line card was taken with one igniter in use; the full-line card, with two igniters. With two igniters the combustion is seen to occur more rapidly than with one, giving a maximum pressure of 335 pounds, as against 272 pounds with one igniter. The difference between the areas of the cards is about 4 per cent.

Gas Cleaners. Most of the large gas engines in use at the present day are running on blast-furnace gas. The success and security of the operation of these engines depend more on the effective cleaning of the gas than on any other single factor. The cleaning is generally carried out by passing the gas through a series of large cooling towers provided with a number of shelves, in which it meets sprays of water. The gas is then taken to a centrifugal fan, which also is supplied with water. The gas and water are thrown against the casing of the fan, and the dust is more or less completely retained by the water.

One of the most effective of these gas cleaners is shown in Fig. 70. The cylindrical drum *E E* is rotated at a speed of about 850 revolutions per minute; it carries on its periphery an oblique vane forming a continuous spiral curve. This vane, together with the casing which it almost touches, makes a spiral channel through which the gas must pass on its way through the cleaner. The front part of the drum, throwing the gas outward by centrifugal force, acts as a suction fan, drawing the gas through the apparatus. Cleaning water enters through tangential openings *B* in the side of the casing, flows through the cleaner in the opposite direction to the gas, and escapes at *I*.

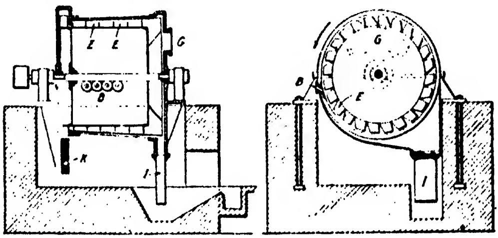


Fig. 70. Sectional View of Theisen Gas Cleaner

The casing is covered on the inside with wire netting, which retains the water. The gas preferably goes to the cleaner when still hot, as this vaporizes some of the water and moistens the dust particles, making them heavier.

HIGH-SPEED ENGINES

Automobile Engines

General Characteristics. Automobile engines are generally vertical multicylinder four-cycle engines designed to run at speeds of 800 revolutions per minute or over, with short strokes, jump-spark ignition, mechanically operated inlet valves, using gasoline as a fuel, and developing not more than 15 horsepower per cylinder at 800 revolutions. The power is usually controlled by throttling with hand adjustment.

The horizontal arrangement is used sometimes with two opposed cylinders—that is, horizontal cylinders lying on opposite sides of the

crankshaft and with their cranks at 180°. Two-cycle engines are also used occasionally, but, so far, have not met with much favor in automobile practice. The standard practice is to use either four or six cylinders arranged in a vertical row, and usually with the cylinders

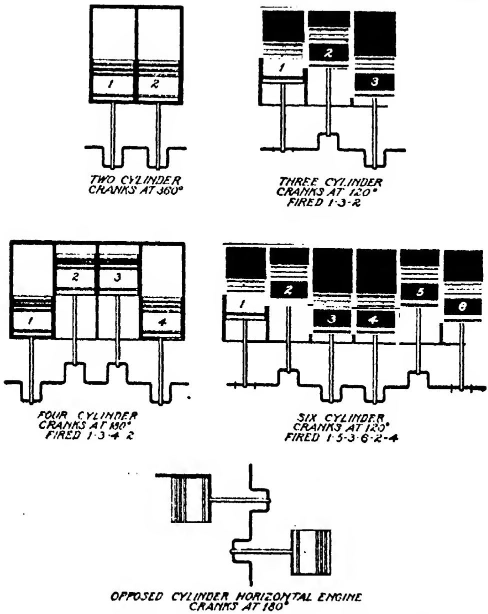


Fig. 71. Arrangement of Multicylinder Four-Cycle Engine for Best Balancing

cast in pairs. Although cylinders cast in triplets and "en bloc"—all the cylinders of the motor cast integrally—are common.

With four cylinders an explosion is obtained twice every revolution of the crankshaft, so that the motor strokes come as frequently as in a double-acting steam engine; with six cylinders there are three explosions per revolution. The explosions are made to take place in

that order which gives the best balancing to the engine—that is, which will give greatest freedom from vibration of the engine as a whole. With a four-cylinder engine, the cranks of cylinders 1 and 4 should be together and at 180° to the cranks of cylinders 2 and 3; the order of firing in the four cylinders should be 1, 3, 4, 2. This is shown in the diagram, Fig. 71, together with the best arrangement of cranks and order of firing for other multicylinder four-cycle engines.

Classification According to Valve Arrangement. There are several standard arrangements of the poppet valves in automobile engines. The classification of the various arrangements is as follows:

The two valves may be on one side of the cylinder—"L" Head Motor.

On opposite sides of the cylinder—"T" Head Motor.

In the head, or else one on the side and one in the head—Overhead-Valve Motor.

The following examples are representative of the various types of motors and show the best present-day practice.

Four-Cylinder "L" Head Motor. This motor, shown in Fig. 72, is a four-cylinder "en bloc" "L" head motor. The cylinders, valve chamber, inlet manifold, water jacket, and cylinder heads are cast integral, the cylinder heads being reinforced by cross ribs. The cylinders and valve passages are entirely water-jacketed, and a drain cock, situated at the lowest level, permits complete drainage. The water inlet is situated directly underneath the exhaust valves. The top plate cover for the cylinder-head jacket, is of bronze or aluminum and forms the outlet through which the water passes to the radiator. This opening on top of the cylinders greatly facilitates the production of perfect and uniform castings, and also the removal of all core sand. The valve covers are screwed into the top of the cylinder with an asbestos-filled copper gasket on the seat of the cap.

The pistons are made of the same grade and quality of cast iron as the cylinders and are slightly tapered. Each piston is fitted with rings of semi-steel, and is provided with four oil grooves. The piston rings are ground on the two sides and fit accurately into the grooves. The wrist pin is of hollow tool steel, ground and hardened, and is fastened to the connecting rod by means of a bolt. This gives a wider bearing surface, because the bearing is in the piston bosses instead of in the connecting rod.

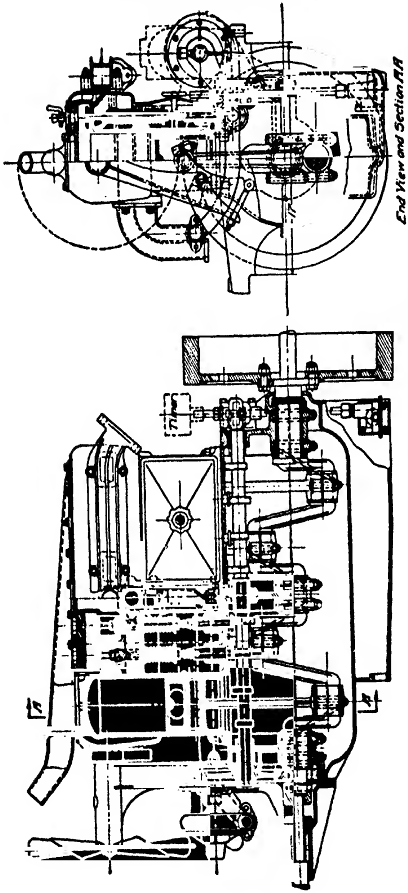


Fig. 72. Typical "L" Head 4-Cylinder Automobile Engine
Courtesy of Wisconsin Motor Manufacturing Company, Milwaukee, Wisconsin

The inlet and exhaust valves are interchangeable. The valve-rod guides are of cast iron, and are very long, thus preventing the gas from escaping when the exhaust valves are open, and also preventing the incoming charge from being diluted by the suction of air.

The valves are cooled by bringing the jacket as close as possible to the valve seats. The cooling is also aided by placing the water inlet directly beneath the exhaust valves. The inlet manifold is cast integral with the cylinders, and is separated from them by the water jacket. The exhaust manifold, made of malleable iron having a gradual taper and provided with fins to aid in the cooling, is bolted to the side of the cylinder casting.

The upper half of the crankcase is a single aluminum casting reinforced by cross ribs. Crankshaft bearings are supported by webs

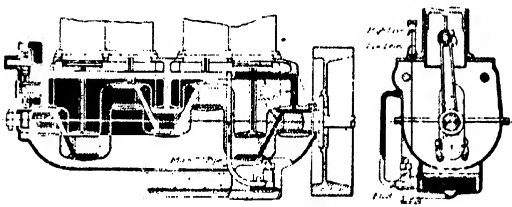


Fig. 73. Oiling System of Wisconsin Motor
Courtesy of Wisconsin Motor Manufacturing Company, Milwaukee, Wisconsin

extending through the entire depth of the case and are held in place by through bolts, which are entirely independent of the lower case.

Oil is pumped by means of a gear pump located on the outside of the lower crankcase and driven by a pair of gears from the camshaft. As shown in Fig. 73, the oil is forced to a main duct cast integral with the crankcase, and from there distributed by means of smaller ducts to the main bearings. It goes through ducts in the crankshaft to the connecting-rod bearings, from which a sufficient amount of oil is thrown off to oil the pistons and camshaft, both of which are provided with oil pockets.

Four-Cylinder "T" Head Motor. In Fig. 74 is shown a transverse section through one cylinder of a four-cylinder "T" head motor, with the cylinders cast in pairs. Each pair of cylinders, the valve chambers, water jacket, and cylinder heads are cast integral. The

top plate cover for each pair of cylinder-head jackets is provided with a flanged opening at its center, which serves as the outlet through which the water passes to the discharge-water manifold, and from there to the radiator. This motor is especially built for truck service,

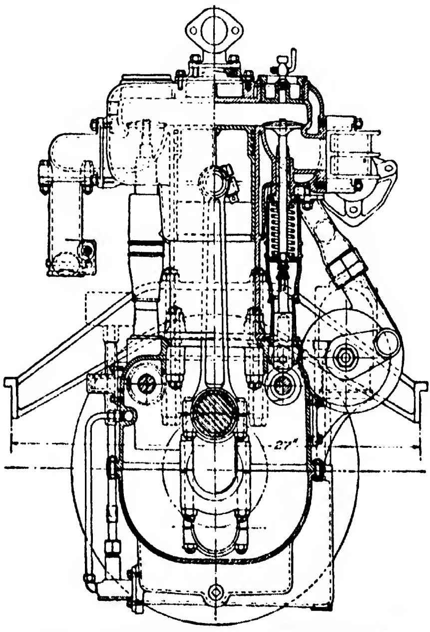


Fig. 74. Transverse Section of Typical "T" Head 4-Cylinder Engine
Courtesy of Wisconsin Motor Manufacturing Company, Milwaukee, Wisconsin

and is fitted with a centrifugal governor acting on a throttle valve in the inlet manifold, so that the truck cannot be driven in excess of the speed at which the governor is set.

The oiling system shown in Fig. 75 differs from that shown in Fig. 73 in several important features. The lower half of the crank-

case is divided into two horizontal compartments, the upper compartment being of pressed steel. The lower compartment carries the oil supply and contains the gear oil pump. The upper compartment contains the oil which is used for the splash oiling, the amount of oil being carried automatically at the same level at all times by means of standpipes of fixed height which connect the upper and lower compartments. The oil pump handles more oil all the time than can be consumed in the proper lubrication of the parts, the excess oil returning from the upper compartment through the standpipes to the lower. The main bearings, the camshaft gears, and the walls of each cylinder are oiled through ducts, as in the previous system, directly from the pump. The connecting rod and camshaft bearings, and the cylinder

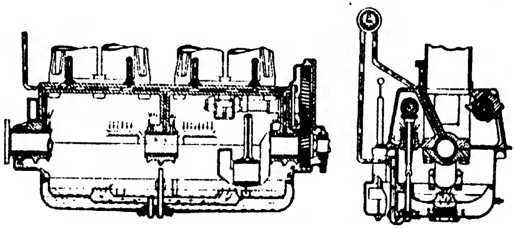


Fig. 75. Oiling System of National "40" Motor
Courtesy of National Motor Vehicle Company, Indianapolis, Indiana

walls to some extent, are oiled by splash from the connecting rods. These rods carry dippers on their lower ends which dip into the oil in the upper compartment.

Overhead Valve Motor. When the valves are placed on top, it is necessary to use levers between the push rods and the valves, with some such arrangement as that shown in Fig. 76. The engine shown in Fig. 77 has both valves in the head. The valves are operated by push rods through rocker arms fulcrumed on the cylinder head. This motor is also noteworthy from the fact that it is air-cooled. The cylinders are cast integrally of vanadium iron with radial fins running lengthwise of the cylinder. Each cylinder is jacketed by means of a band of sheet iron around the outside of the fins, and the upper part of of the engine is shut off from the lower part by a horizontal bulkhead, so that the only communication between the upper and the lower

compartments is through the space between the cylinder walls and the jacket around the outside of the fins. Thus, if a current of air is caused to flow from the upper to the lower compartment, the fins are cooled by the air current. When placed in the chassis, the engine is located in an air chamber formed by the engine boot and hood, the only exit from which is through the flywheel, and the only entrance to which is through the area around each cylinder. A "sirocco" suction fan is built into the flywheel, and when the flywheel rotates, a partial vacuum is created in the lower compartment. Large quantities of air are then forced by atmospheric pressure in, down, and around each cylinder to take the place of the air exhausted by the fan.

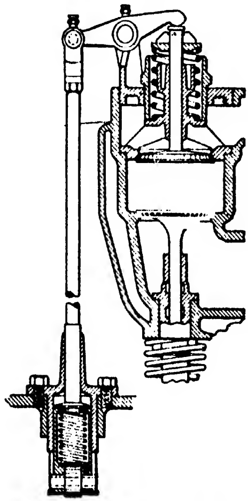


Fig. 70. Section Through Valves, Overhead Valve Automobile Engine

Silent Knight Sleeve-Valve Motor. Previous to 1908, the only form of valve that had proved commercially successful on internal-combustion motors—with the exception of valveless motors in which the piston acts as a valve—was the poppet valve. Slide valves and all the different forms of valves used in steam engines, including flat-slide, piston, and rotary valves, were at one time or another applied to internal-combustion engines, but never in the long run proved commercially successful. In 1908, Charles Y. Knight, of Chicago, induced the Daimler Motor Company of Coventry, England, to take up a motor of his invention in which the valve functions are performed by two ported sliding sleeves, one within the other, and both between the piston and the cylinder wall. The two sleeves are reciprocated by a half-speed shaft carrying small cranks equal in number to the sleeves. This motor has since proved very successful, and has been taken up by

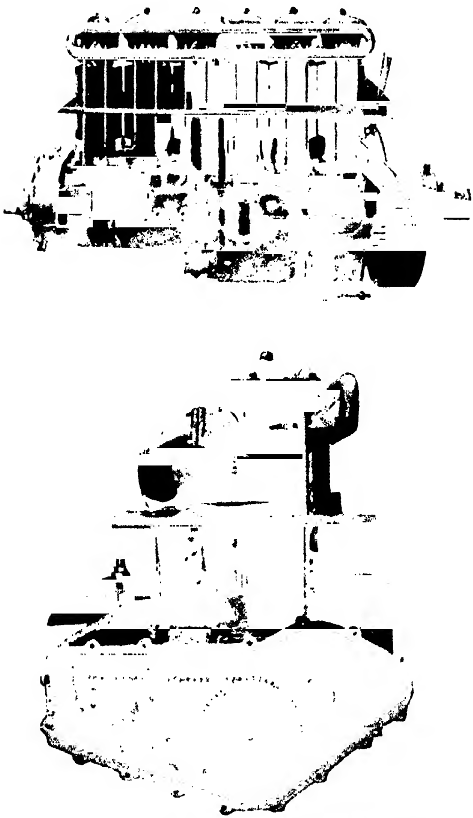


Fig. 77 Side and End Views of Franklin Air-Cooled, Overhead Valve Motor
Courtesy of H. H. Franklin Manufacturing Company, Syracuse, New York

leading automobile manufacturers in different European countries and in the United States.

Action of Sleeve Valves. In Figs. 78 and 79 two Knight motors made by different concerns are shown, which differ only in details of design. The Knight motor is a four-cycle gasoline engine in which

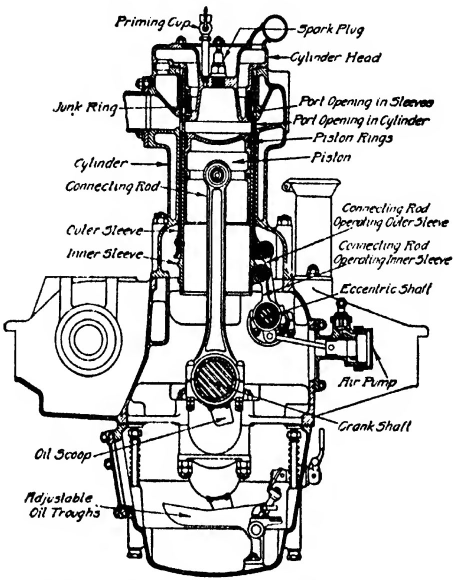


Fig. 78. Transverse Section of Stearns-Knight Sleeve-Valve Motor
Courtesy of F. B. Stearns Company, Cleveland, Ohio

the usual poppet valves have been replaced by two concentric sleeves, which slide up and down between the walls of the piston and cylinder. Each of these sleeves has ports or slots cut on opposite sides. These slots in the inner and outer sleeves register with one another at proper intervals, producing large and direct openings into the combustion chamber from the exhaust and inlet ports. The cylinder head is

detachable and part of it extends into the cylinder for quite a distance. However, it has a smaller diameter than the cylinder bore, so that the two sleeves may enter between it and the cylinder wall. Since there are no valve passages between the valves and the cylinder bore, the combustion chamber is made nearly spherical by making the cylinder head and piston head concave or bowl-shaped. The two sleeves are independently operated by small connecting rods working from an eccentric shaft running lengthwise of the motor. This eccentric shaft is positively driven by a chain at one-half the speed of the crankshaft. The connecting rods are fastened to the sleeves by pins through lugs formed on the lower ends of the sleeves, giving a drive which is unsymmetrical or one-sided, but which has no bad effect, owing to the long bearing of the sleeves. The connecting rods for the two sets of sleeves are of different lengths, and usually they are inclined, i.e., a vertical axis drawn through the center of the eccentric shaft lies outside of the pins connecting the rods to the sleeves. The sleeves are strengthened by circumferential flanges at the lower ends, above and below the driving lugs.

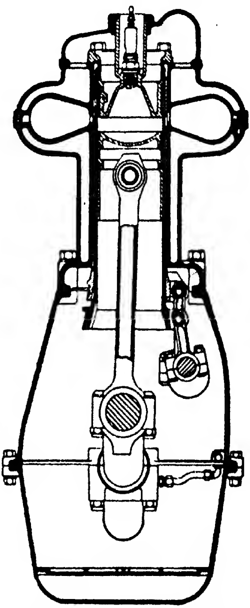


Fig. 79. Transverse Section through Moline-Knight Sleeve-Valve Motor
Courtesy of Moline Automobile Company, East Moline, Illinois

Packing Rings. Gas tightness is insured by two sets of packing rings, one set of three or four being placed on the piston in the usual way, and the other set on that portion of the cylinder head which extends into the cylinder and is surrounded by the valve sleeves. The latter set comprises one unusually broad ring at the bottom

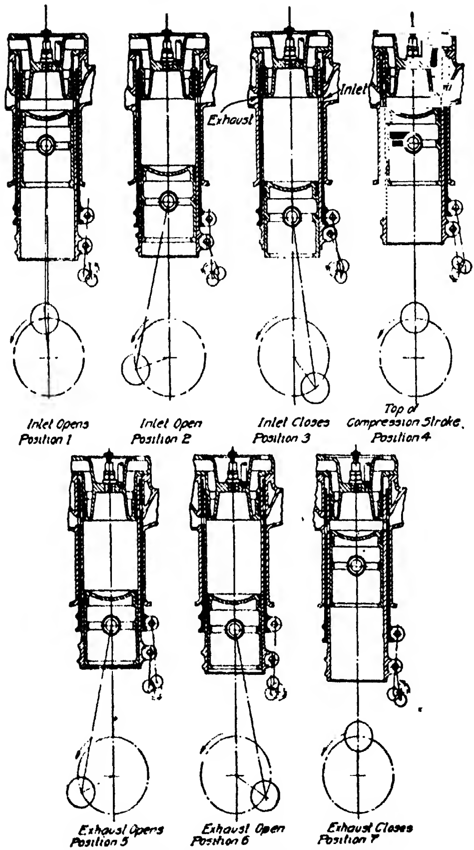


Fig. 80. Valve and Piston Positions for Stearns-Knight Motor
Courtesy of F. B. Stearns Company, Cleveland, Ohio

known as the junk ring, and two or three rings above it of the same dimensions as used on the piston. The object of the junk ring is to seal the ports in the valve sleeves while the explosion takes place and the gases expand in the cylinder. This ring, therefore, must be made of greater width than the height of the ports. Each port extends substantially one-third of the distance around the cylinder and is provided with a bridge at the middle of its length so as to prevent undue weakening of the sleeve.

Position of Sleeves at Various Points in Cycle. The eccentric driving the inner sleeve is given a certain lead or advance over that operating the outer sleeve from 60° to 90°. This lead, together with the rotation of the eccentric shaft at half the crankshaft speed, produces the valve action illustrated in Fig. 80, which shows the relative position of the piston, sleeves, and cylinder ports at various points in the rotation of the crankshaft.

Position *1* shows the inlet just opening. The port or slot in the inner sleeve is coming up, the port in the outer sleeve is going down, and the passage for the incoming gas is a rapidly increasing space between the upper edge of the slot in the inner sleeve and the lower edge of the slot in the outer sleeve.

Position *2* shows the inlet full open. Both slots have come exactly opposite each other and the mouth of the inlet port in the cylinder.

Position *3* shows the closing of the inlet. The cylinder is now charged with gas and ready for the compression stroke.

Position *4* shows the position of the sleeves at the top of the compression stroke; the compression space in the cylinder is completely sealed by the rings in the head and the rings in the piston. The explosion takes place at this point.

Position *5* shows the exhaust valve just starting to open. The slot in the outer sleeve is coming up and the slot in the inner sleeve is going down.

Position *6* shows the exhaust full open. Both slots have come opposite each other and the exhaust port in the cylinder.

Position *7* shows the closing of the exhaust opening, and is practically identical with Position *1*. The four cycles or strokes of the engine, i.e., suction, compression, explosion, and exhaust, have now been completed; the crankshaft has turned twice, the eccentric shaft

has driven the sleeves up and down once, and the cycle of operation is now ready to be repeated.

Timing. The timing shown is not different from that ordinarily used in poppet-valve engines, but the valve area is greater than that of an ordinary poppet valve. The equivalent of increased valve area is gained, also, by the directness of the valve opening and the absence of restrictions in the gas passages.

Advantages of Knight Construction. The two chief advantages claimed for the Knight motor are, first, its high output for given cylinder dimensions, which is due to the directness of the path of air into the engine; large area and rapid opening and closing of the valve ports; absence of heating of the charge by passing over heated valves and valve pockets; and to the favorable form of the compression chamber, with the spark plug located in the center of the head. The second advantage is the silence in operation, and the fact that the valve timing and the efficiency of valve operation are not affected by running for long periods, in fact, it is found that the compression of a new engine is improved by running, due to carbon filling up inequalities made in machining the sleeves, cylinder, and head. Another advantage is the small jacket loss due to a combustion chamber with a minimum wall area and a minimum exhaust-wall area.

The silence in operation at all speeds is due to the fact that the valves are closed as well as opened positively.

If great power and high speeds are desired in the poppet-valve motor, high compression, large valves, strong springs, and steep cams are employed, producing a noisy motor.

Large valves and their seats are liable to warp or to be pounded out of shape under the combined action of high temperature and the impact from strong springs. The pressure exerted by the spring sometimes reaches 300 pounds.

With the sleeve valve, the efficiency and durability of the engine are not affected by high pressures. The sleeve valve is balanced against lateral pressure, and the explosion does not affect or shock it at any point. The ports are large, and the action of the motor is not affected adversely by their large size.

Most of the time, during which the pressure in the cylinder is considerable, the sleeves travel in the same direction as the piston, thereby minimizing the power required for moving the sleeves. Fig.

81 shows that both sleeves move up with the piston during the compression stroke, and that the inner sleeve moves down with the piston during the power stroke. The outer sleeve, however, moves in opposition to the piston during the power stroke. Mr. Knight states that tests made on a six-cylinder 75-horsepower motor showed that 2 horsepower was required for driving the half-speed shaft.

Lubrication. The early Knight motors depended entirely upon splash lubrication for their oiling. The engine shown in Fig. 78, the oiling system of which is shown in Fig. 82, has a combined splash- and pressure-feed-lubrication system. Crankshaft bearings, pump-

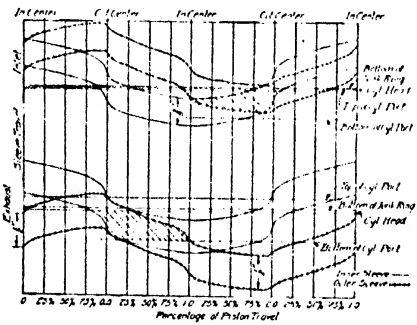


Fig. 81. Valve Action of Knight Motor, Showing Effective Port Opening for Different Piston Positions

shaft bearings, and the silent chains driving the eccentric shafts, are fed from the pressure system. Dippers secured to the heads of the connecting rods splash into oil troughs located under the connecting rods and form a mist of oil which is thrown on to the connecting-rod and wrist-pin bearings, the pistons, sleeves and sleeve connecting rods, and eccentric shaft bearings. The troughs are so arranged that they are raised and lowered as the throttle is opened and closed, respectively, so that the higher the speed the deeper the dippers on the connecting rods splash below the surface of the oil in the troughs. The motor shown in Fig. 79 is lubricated by a pressure-feed system. The flow of oil delivered under pressure is determined by a valve

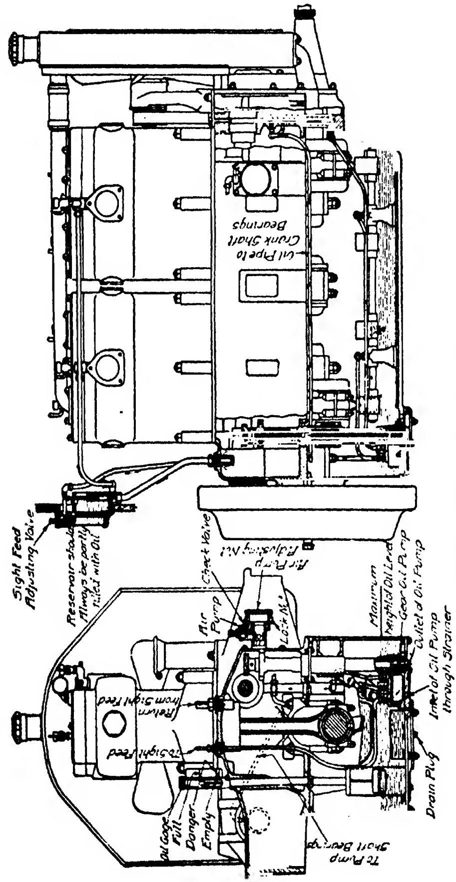


Fig 82 Oiling System of Stearns-Knight Motor
Courtesy of F B Stearns Company, Cleveland, Ohio

which is so connected as to open and close with the throttle. There are no oil grooves in any of the crankshaft bearings.

The motors depending upon splash lubrication for the sleeves have numerous oil holes drilled in the pistons, so that the splash may pass through to the sleeves. All Knight motors, as made in this country, have oil holes in the sleeves, so that oil may pass from the inner to the outer sleeve, and from there to the cylinder wall. This is the case whether the lubrication system is splash or pressure. The sleeves, in every case, have oil grooves cut on the outer surface

Fig. 83. Sleeves of Moline-Knight Motor
Courtesy of Moline Automobile Company, East Moline, Illinois

by which the oil is distributed around the circumference. The forms of the inner and outer sleeves, showing the various systems of oil-grooving, are shown in Fig. 83.

The oil works between the sleeves, between the outer sleeve and the cylinder wall, and between the inner sleeve and the piston, aided by the suction in the ports during the inlet stroke. The upper end of the sleeves, above the ports, are lubricated by the suction on top of the sleeves between the cylinder wall and head, due to the downward movement of the sleeves during the exhaust and the first part of the suction strokes.

Fig. 84 shows a part sectional view of a Knight motor, having one cylinder intact, the next with the cylinder wall cut away to show the outer sleeve, the third with the outer sleeve cut away showing the

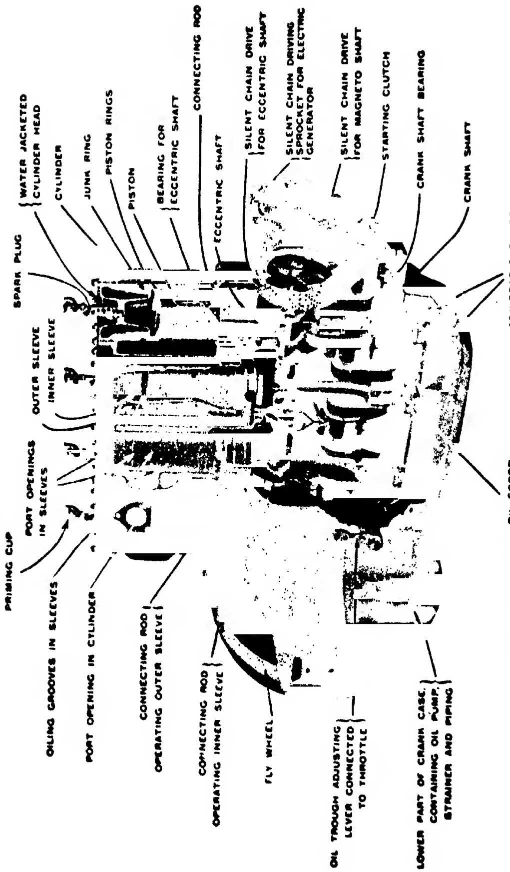


Fig. 84. Part Section of Stearns-Knight Motor, Showing Sleeve Valves
Courtesy of F. B. Stearns Company, Cleveland, Ohio

inner, and the last with the inner sleeve cut away showing the piston and the cylinder head cut on the center line showing its cross section.

Automobile Motor Rating. The formula for the brake horsepower (b.h.p.) of an engine has been given in the section on "Thermodynamics". Several empirical formulas have been devised for small high-speed engines, which give fairly accurate results if the engine under consideration is of the same dimensions, has the same shape of combustion chamber, and operates under the same conditions as the engines from which the empirical constants were obtained.

Four-Cycle formulas:

Institution of Automobile Engineers (Great Britain) formula

$$\text{b.h.p} = 0.45\,(D - 1.18)\,(L + D)$$

A.L.A.M rating formula

$$\text{b.h.p.} = \frac{D^2 N}{2.5}$$

Roberts' formula

$$\text{b.h.p.} = \frac{D^2 L N R}{18{,}000}$$

Royal Auto Club formula

$$\text{b.h.p.} = \frac{(D + L)^2 N}{9.92}$$

In these equations, D is the piston diameter in inches; L is the stroke in inches; R is the revolutions per minute; and N the number of cylinders.

The first of these formulas is intended to give the maximum horsepower which can safely be obtained from an engine of given cylinder dimensions under the most favorable conditions; the second is intended to give the power when the motor is running with a piston speed of 1000 feet per minute, a nominal speed which can be exceeded by most motors. The first formula gives the highest results, the second the lowest, the third gives results which very closely approximate the actual, and the fourth is an average between the first and second

Racing-Boat Formulas. The following formulas for high-speed racing-boat engines of the four-cycle type, are based on 1000 feet per minute piston speed. For engines of ordinary design, two-thirds of the above values should be taken; 10 per cent should be added to the ratings if the charge is forced into the cylinders by any mechanical device.

American Power Boat Association

$$\text{h.p.} = \frac{D^2N}{2.5338}$$

For motors of less than 6-inch stroke

$$\text{h.p.} = \frac{D^2LN}{15.20}$$

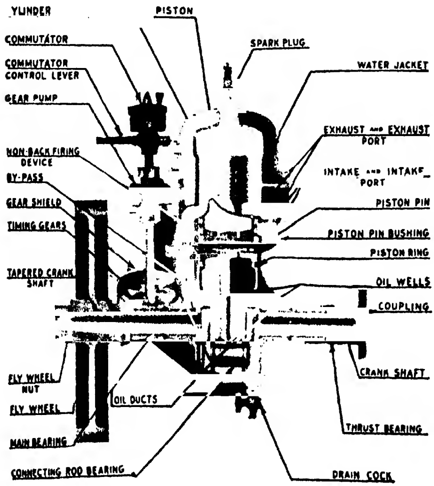


Fig. 85. Section of Two-Cycle Marine Engine
Courtesy of Gray Motor Company, Detroit, Michigan

Two-Cycle Formulas. Three formulas for two-cycle racing-boat engines, the first by Roberts, and the next two by the A. P. B. A., are as follows:

$$\text{h.p.} = \frac{D^2LRN}{13{,}500}$$

$$\text{h.p.} = \frac{D^2N}{2.1008}$$

$$\text{h.p.} = \frac{D^2LN}{12.987}$$

The preceding formulas by the American Power Boat Association are only for racing-boat engines. For ordinary two-cycle boat engines, two-thirds of the value resulting from the use of these formulas should be taken.

Marine Engines

Increased Use of Two-Cycle Motors. The principal difference between marine and automobile practice is in the much more extended

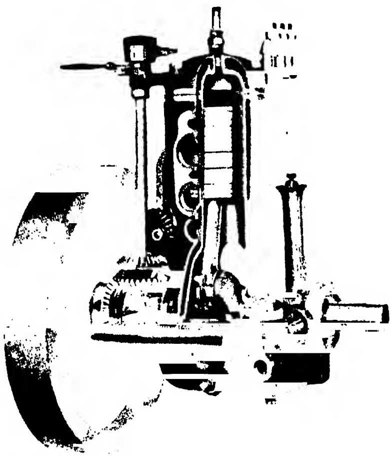


Fig. 86. Ferro Two-Cycle Marine Motor
Courtesy of Ferro Machine and Foundry Company, Cleveland, Ohio

use of two-cycle engines for small powers in motor boats. Where four-cycle engines are used, they do not differ appreciably from automobile engines, except that they are very generally made stronger and heavier, and often of larger size and lower speed. The use of two-cycle engines is more prevalent in marine practice than in stationary practice, because of the fact that the increased power for a given weight of a two-cycle motor is of more importance than the increased fuel consumption. The motors shown in Figs. 85, 86, and 87 are most frequently used in pleasure boats.

Motor with Extra Ports. The motor shown in Fig. 85 has a second port controlled by a valve which is valuable for giving good control at low speed and which makes for easy starting, and a third port which is uncovered near the top of the piston stroke, which increases the pressure of the charge in the cylinder and consequently is

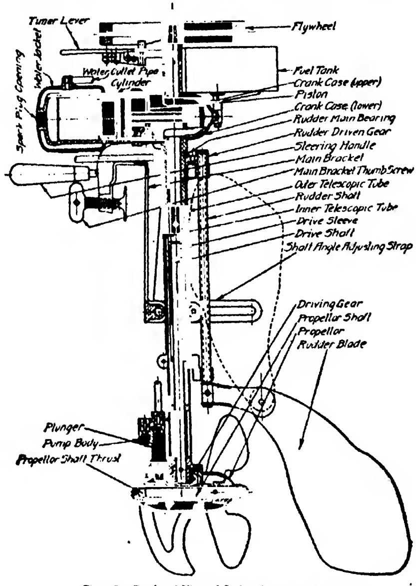


Fig. 87. Sectional View of Caillie Portable Motor
Courtesy of Caillie Perfection Motor Company, Detroit, Michigan

valuable in increasing the power, particularly at high speeds. The by-pass plate in the transfer port carries a device to prevent back-firing in the crankcase; it works on the same principle as the miner's lamp.

Rowboat, or Portable, Motor. The motor shown in Fig. 87 is a recent development for rowboats. The horizontal flywheel in this type of motor has a gyroscopic effect and aids in steadying the boat.

Aeronautical Motors

Characteristics of Light Weight and High Reliability. This type of motor differs from other high-speed motors in that the weight for the power developed must be a minimum, while the reliability must be a maximum. The cost of manufacture is relatively unimportant, consequently many refinements of construction are available which could not be considered for the other high-speed engines. For instance, pistons are finished all over, cylinders often machined from a bar of solid steel, and the connecting rods drilled out to reduce the weight as far as possible. In addition to this, extremely short strokes and a saving in crankcase and crankshaft weight by so arranging the cylinders that the same crankshaft and crankcase serve two or more rows of cylinders and reduce the weight to as low as 1.75 pounds per horsepower.

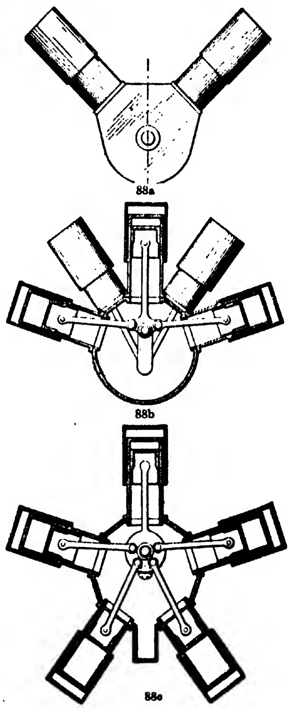


Fig. 88. Typical Cylinder Arrangements for Aeronautical Motors

Arrangement of Cylinders. The most usual arrangements of the cylinders are shown in Fig. 88, where (a) is the V-type, (b) the fan type, and (c) the radial or star type. The V-type is the most common, the number of cylinders running as high as twenty-four. An example of an eight-cylinder Curtiss motor of this type is shown in Fig. 89. The radial

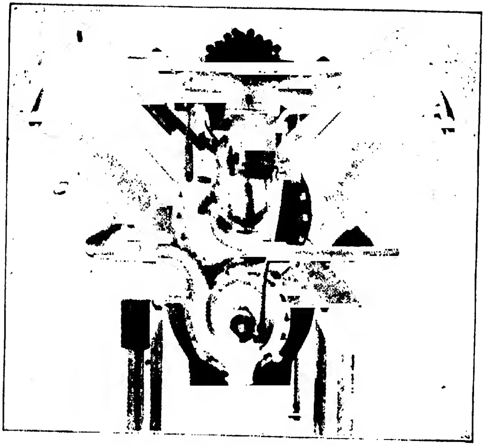


Fig. 89. Curtiss 90-100 H. P. Model O-X Motor
Courtesy of Curtiss Motor Company, Hammondsport, New York

Fig. 90. Gnome Seven-Cylinder Revolving Motor

type may have either the cylinders stationary and the crankshaft revolving, or the crankshaft stationary and the cylinders revolving. In the latter case a heavy flywheel effect is obtained without the use of a flywheel. The best-known example of this type is the Gnome motor, shown in Fig. 90, a motor which has probably been more widely used for aviation work than any other. The rapid rotation of the cylinders and the use of fins on their outsides makes adequate air-cooling easy.

LOW-PRESSURE OIL ENGINES

Characteristics. The field for the use of the oil engine is very extended. It is the most compact of the heat engines, requiring nothing equivalent to boiler or gas generator, and consequently is inherently the most suitable for purposes of transportation. Its extensive adoption in stationary plants is being followed by its application to locomotives and to large vessels. The absence of boiler and of gas-generator losses makes it both potentially and actually the most efficient of all heat engines.

Oil engines in which the fuel is gasified in external vaporizers do not differ from gas engines as far as the engine itself is concerned.

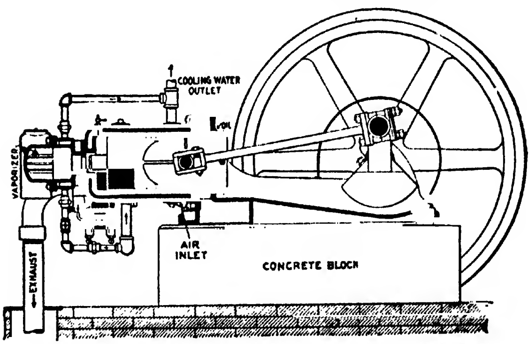


Fig. 91. Longitudinal Section of Hornsby-Akroyd Oil Engine
Courtesy of De La Vergne Machine Company, New York City

Any gas engine may be fitted with an external vaporizer if the compression is not so high as to pre-ignite the charge. Consequently, only those engines in which the fuel is vaporized within the cylinder will be here described.

De La Vergne Type H. A. Oil Engines. Figs. 91 and 92 show longitudinal and transverse sections, respectively, of this engine.

Vaporizing Apparatus. The vaporizer in this engine has been schematically shown in Fig. 37*a* and described on page 83. The vaporizer consists of two parts—the vaporizer jacket and the vaporizer cap. The vaporizer jacket is bolted to the cylinder head and

connects with the cylinder through a narrow passage; the vaporizer cap consists of an unjacketed gun-iron thimble, heavily ribbed on the inside to increase its surface, and is bolted to the vaporizer jacket. The fuel is generally stored in a tank outside the powerhouse, from which it is raised by a small rotary pump, driven by the engine, to a standpipe located near the engine base. The pump keeps the standpipe filled, and an overflow pipe carries the surplus back to the reservoir. The oil is withdrawn from the standpipe by the oil pump shown in the transverse section, Fig. 92, and injected into the vaporizer during the suction stroke. The pump is actuated by the inlet-valve lever, so that the two operations are simultaneous.

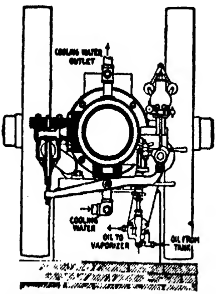


Fig. 92. Cross Section of Hornsby-Akroyd Oil Engine
Courtesy of De La Vergne Machine Company, New York City

Regulation. The regulation is secured by varying the amount of oil injected in proportion to the load. The governor lever controls the double by-pass valve shown in section in Fig. 93, and separates the oil handled by the pump into two parts, one of which enters the vaporizer, while the other flows back to the reservoir—the returning oil may be seen through the gage glass in the return

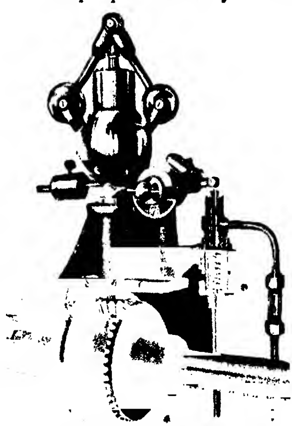


Fig. 93. Governor and By-Pass Valve
Courtesy of De La Vergne Machine Company, New York City

pipe. When operating under ordinary conditions of load, the governor opens only the small inside valve. If the engine should tend to speed up, both the small inside valve and the large concentric outer valve will be opened and all of the oil will be by-passed and none of it will be injected. This arrangement prevents any possibility of the engine running away.

These engines operate successfully on kerosene, any of the crude oils produced in the Eastern and Middle States, or nearly all of the distillates and residual fuel oils 24° Baumé or lighter.

The abrupt contraction of the vaporizer neck prevents any carbon which may be deposited in the vaporizer from getting into the cylinder. The construction of the vaporizer cap permits of easy removal and cleaning, which should be done at intervals of approximately a week, depending upon the quality of the fuel and the load carried.

Mietz and Weiss Two-Cycle Type. Another low-pressure oil engine of a slightly different type is shown in Fig. 38 and described on page 85. As this is a two-cycle engine, air is taken into the closed crankcase from the interior of the base through the suction port when that is uncovered by the piston, near the end of the compression stroke. The air enters the cylinder (after slight precompression in the crankcase) through the air port in the way usual with two-cycle engines. It carries with it whatever steam has been formed in the jacket, the steam coming from the chamber *E* through the pipe *F* to the transfer port. The water level in the jacket is kept constant by a float. The exhaust occurs through the port *G* at the same time as the admission of the charge through the transfer port, the deflector plate on the piston preventing the charge from blowing directly across the cylinder and out of the exhaust port. The oil is injected during the compression stroke onto the projecting lip of the hot bulb *C*, instead of into it, as in the engine previously described.

Mietz and Weiss Three-Cylinder Engine. Fig. 94 gives a section of a three-cylinder vertical engine of this type. The oil distribution to the cylinders is obtained as follows: The oil pump is driven from a shaft which runs as many times faster than the crankshaft as there are cylinders, and is so set that each down stroke of the plunger coincides with the upstroke of each piston in turn. The oil is delivered from the pump to a distributor, which runs at the same

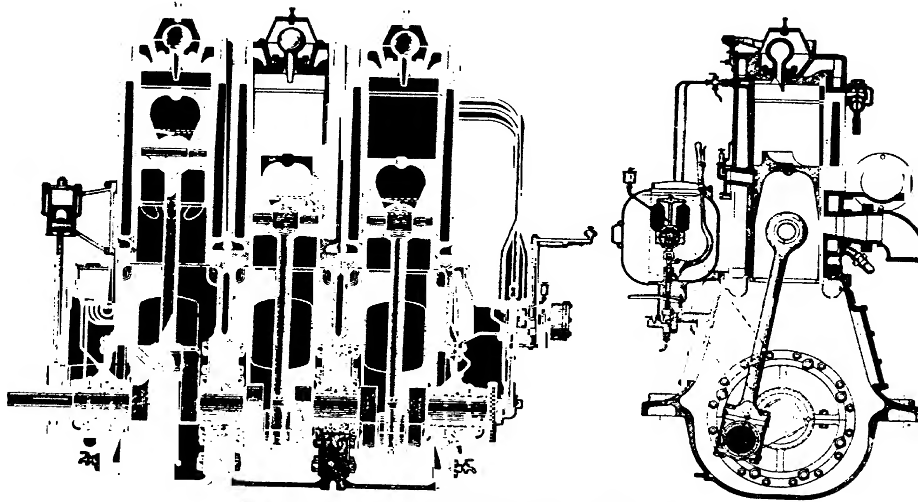


Fig. 94. Longitudinal and Transverse Sections of Mietz and Weiss Oil Engine
Courtesy of August Mietz, New York City

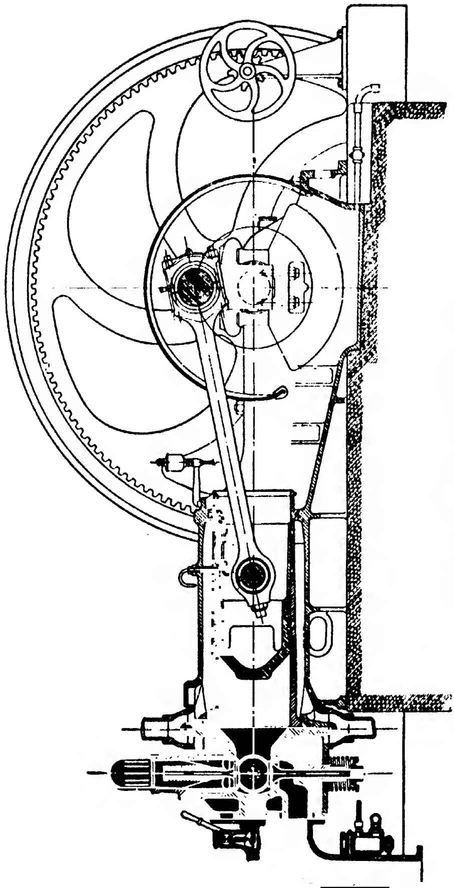


Fig. 93. Sectional Elevation of New Crossley Oil Engine
Courtesy of Crossley Brothers, Manchester, England

speed as the engine shaft. This distributor consists of a disk with one port, through which the oil passes on its way from the pump to the cylinders, and a case which has as many outlets for oil pipes as there are cylinders. The distributor disk is so set that it uncovers in turn the oil outlet to the cylinder whose piston is on the upstroke while the pump plunger is on its down stroke.

Crossley Oil Engine. An English oil engine with internal vaporizer is shown in Fig. 95 and a cross section through the combustion chamber and vaporizer is given in Fig. 96. The vaporizer consists of a loose cover secured in place by a circular ring. It is fitted with an ignition tube, by the aid of which the starting of the engine can be effected in about five minutes. The oil sprayer is fitted in a water-cooled portion of the breech end, opposite to the vaporizer. The fuel is injected at about the end of the compression stroke in the form of finely divided spray. Part of this passes through the hot compressed air in the combustion chamber and impinges on the vaporizer. Ignition takes place as the crank passes the inner center, and the piston is then driven out on the power stroke.

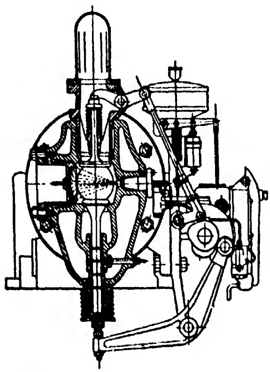


Fig. 96. Cross Section through Combustion Chamber and Vaporizer of Crossley Engine

Oil Pump. The oil pump, Fig. 97, consists of a bronze body with steel valves and fittings. It is worked by means of a steel lever, which, in turn, is operated by means of a cam on the side shaft. The pump, pump lever, and cam work in an oil bath to ensure thorough lubrication and quiet action.

Governing. The governing of the engine, Fig. 97, is effected by means of a system which varies the time at which a control valve is opened, and through which any oil not required returns to the suction side of the oil pump. The control valve is opened by a wedge

being interposed between the end of the control-valve spindle and an arm on the oil-pump lever. The wedge is raised or lowered by a centrifugal governor, and the time when the control valve is opened depends on the position of the wedge, the latter in turn depending on the speed of the engine. An impulse takes place on each cycle, and the power of the impulse is graduated according to the speed and the load on the engine.

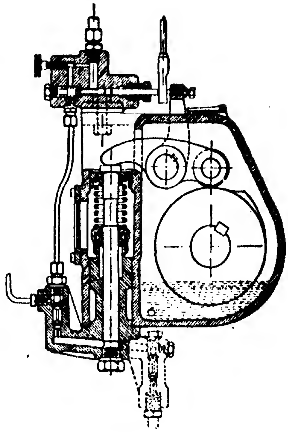


Fig. 97. Oil Pump and Governing Gear of Crossley Engine

The important feature of this gear is that the oil pump, which has a constant length of stroke, always begins to deliver the oil through the oil sprayer at the same time and at the same speed, and therefore the spraying is equally efficient at all loads. Also, the actual governing effect takes place at the moment the oil fuel is being injected and burned.

When working on lighter loads, the wedge comes into action sooner, with the result that soon after the pump begins to deliver oil the control valve opens, the injection of oil through the sprayer suddenly ceases, and the pump delivers the remainder of the oil through the control valve back to the suction side of the pump.

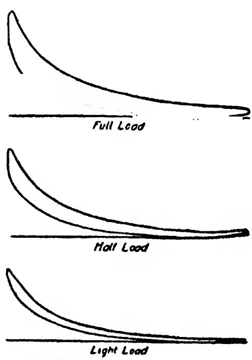


Fig. 98. Typical Indicator Cards for Crossley Oil Engine

Indicator Cards. Fig. 98 shows indicator cards obtained from this engine. The method of governing gives good efficiency, not only at the maximum load, but also at reduced loads. In a recent test the oil consumption per brake horsepower per hour did not vary more than 5 per cent above .44 pound (.38 pint or 200 grammes) from maximum load to half-load.

An oil heater is used with the more viscous oils for the purpose of heating the oil sufficiently to make it flow easily through the pipes and pump. The heater is secured to and heated by the exhaust pipe.

DIESEL OIL ENGINES

Diesel engines, with the exception of the fuel atomizers, oil pumps, and injection air compressors, are similar to gas engines in details of construction, except that they are built heavier to withstand the higher pressures obtaining with this cycle. The Diesel cycle can be carried out either in four strokes or in two strokes.

STATIONARY DIESEL ENGINES

Four-Cycle Type. *Busch-Sulzer Engine.* In the engine shown in Fig. 99, the valves are located, one over the other, in a valve chamber to the side of the cylinder. The shape of the clearance space resulting from this location of the valves, together with a flat piston and cylinder head, is not as favorable to the best combustion results as a spherical shape would be. The atomizer is horizontal and is shown in Fig. 39, and described on page 88. The high-pressure-injection air is furnished by a separate two- or three-stage compressor, belt-driven from the engine shaft or motor-driven as desired. The engine is air-started by means of a special starting valve.

Fulton-Tosi Engine. The engine shown in part in Fig. 100 has an "A" frame, which forms not only the frame but also the cylinder jacket. The cylinder wall is formed by a liner set into the top of the frame, thus making it possible to use two grades of iron especially suitable for the purposes to which they are put—a soft iron for the frame, to insure a good casting; and a hard, close-grained iron for the liner, to withstand the wear. This construction also makes it possible to replace the cylinder wearing-surface without scrapping a whole cylinder, and also permits of longitudinal expansion of the liner to correspond to the unequal heating of the jacket wall and cylinder

liner. Both the exhaust and inlet valves are located in the cylinder head. With this construction the head can be dished and, together with a dished piston head, forms a combustion chamber which is nearly spherical in shape. The atomizer, Fig. 40, is located in a vertical position in the center of the head; this, with the spherical-shaped combustion chamber, tends to promote the most efficient combustion.

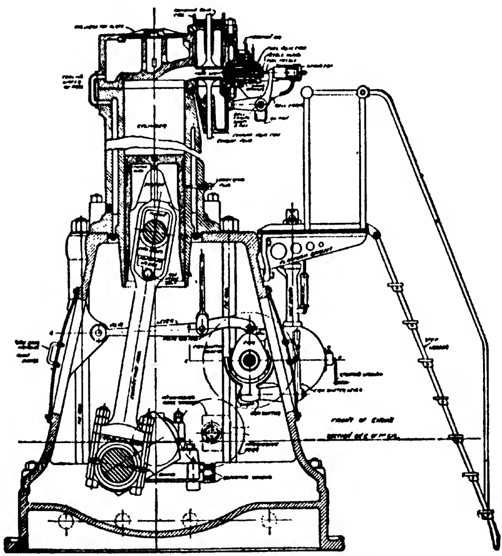


Fig. 99. Sectional View of a Diesel Oil Engine
*Courtesy of Busch-Sulzer Brothers-Diesel Engine Company, St. Louis

The fuel-valve lever is mounted on an eccentric pin; by rotating this pin to various positions, the lead and length of opening of the fuel valve may be varied while the engine is in operation to secure quiet and economical operation at various loads. Each cylinder is provided with an air-operated starting valve to which air is admitted by a rotary distributor driven by the camshaft. In the larger sizes, the heads of the

pistons are water-cooled—the water being admitted and discharged from this jacket by telescope tubes, as shown in Fig. 100. The cylinder is oiled by a force feed pump delivering oil at four points around the bore, the timing of the discharge and the level of the points of intro-

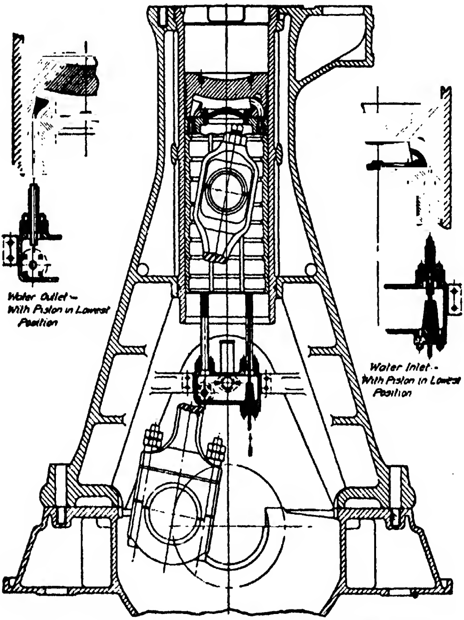


Fig 100 Fulton-Tosi Diesel Engine, Showing Cooling-Water Arrangement as Applied to Piston of Large Engines
Courtesy of Fulton Iron Works, St. Louis, Missouri

duction being such that the oil is forced in between the packing rings as the piston approaches the lower end of the stroke. Two-stage compressors are used on the smaller sizes of engines and three-stage on the larger. An inter-cooler between the stages and an after-cooler

for the discharge from the high-pressure stage are provided. The low-pressure intake valve is governor-regulated so that only the amount of air called for by the load is compressed, thus relieving the compression of extra work.

Snow Engine. The engine shown in longitudinal sectional view in Fig. 101 is a horizontal single-acting four-cycle Diesel engine. The jacket walls of the cylinder are formed by the end of the frame casting, and the cylinder wall is formed by a removable liner placed within this casting. Guides for a crosshead are provided in the frame so that the cylinder liner is relieved of the wear due to the thrust of the connecting rod.

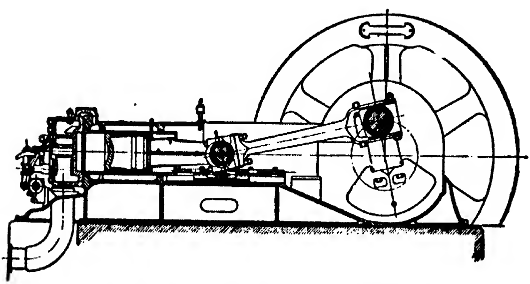


Fig. 101. Longitudinal Section of Snow Four-Cycle Oil Engine
Courtesy of Snow Steam Pump Works, Buffalo, New York

Two-Cycle Type. The fact that air, only, is admitted and compressed in a Diesel engine makes it peculiarly adaptable to modification to the two-cycle principle without loss of efficiency. In the Otto two-cycle engine, scavenging is obtained in one of two ways, either one of which leads to a loss of thermal efficiency. In one, the scavenging is obtained by means of the fresh mixture, in which case the scavenging is either incomplete, leading to loss of efficiency because of the presence of the unexpelled exhaust gases in the fresh mixture, or it is complete, with the result that some of the mixture is lost out of the exhaust port, as it is practically impossible to close the exhaust at exactly the right instant. The other means of scavenging is to force the exhaust gases out by a blast of scavenging air before the

admission of the mixture; this leads to inefficiency, since it destroys the proper mixture proportions.

In the Diesel engine, however, since the charge is air alone, the scavenging can be complete without the loss of anything but air to the exhaust. The combustion is, therefore, as complete as in a four-cycle Diesel engine and no fuel is lost to the exhaust. Since the power of a Diesel two-cycle engine is nearly twice the power obtainable from a four-cycle engine of the same dimensions—the power necessary to drive the scavenging pump being the only additional loss—the two-cycle engine is particularly well adapted for use in marine work, where weight and space per unit of power must be saved, or for large stationary engines in cities, where saving of space is of importance.

In all two-cycle Diesel engines, as at present constructed, the exhaust ports, located in the cylinder wall near the bottom of the piston stroke, are uncovered by the piston. This leads to a simplification and saving of weight in the valve gear.

MARINE DIESEL ENGINES

Both four- and two-cycle engines are used for marine service, the two-cycle seemingly being the better suited to the service because of the greater power per unit weight and because of the fact that such an engine is reversible, while a four-cycle engine must have a reversing gear. The principal difference between stationary and marine four-cycle engines is that the marine engines have enclosed box frames which are continuous throughout the length of the engine, instead of "A" frames, as is the more common practice in stationary engines.

Two-Cycle Type. *Carels-Diesel Engine.* The engine shown in Fig. 102 is a four-cylinder reversible two-cycle marine engine provided with crossheads instead of trunk pistons. The bedplate, narrow "A" frames, and the method of carrying the cylinders are in accordance with marine steam-engine practice. The cylinder liner is a separate casting pressed into the cylinder casting, with the exhaust ports cored out so that water from the cylinder jacket can circulate through and keep them cool, as in the half-sectional view. The exhaust chamber, in back of the cylinder-liner exhaust ports, is entirely contained in the cylinder jacket. At the bottom of the liner is a lantern ring, Fig. 103, communicating with a passage which

is connected to the scavenging-pump suction, thus preventing the escape of gases past the piston rings from the combustion space into the engine room. The cylinder head is of somewhat complicated design, to permit of the introduction of the seven necessary valves,

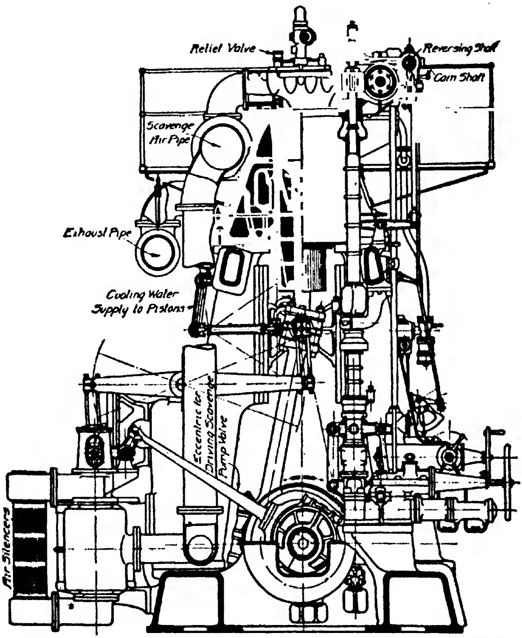


Fig. 102. 1300 I.H.P. Carels Two-Cycle Diesel Engine of Single-Screw Motor Ship "Eavestone"
Courtesy of "The Engineer", London, England

viz, the fuel-injection valve, starting-air valve, safety valve, and four scavenging or inlet valves. The cylinder head is dished upwards and forms, with the dished piston head, a favorable combustion space. The atomizer is similar to that shown in Fig. 40.

The piston, Fig. 103, is in two parts, the upper part being water-jacketed, and the lower portion being an internally ribbed shroud, merely to cover the exhaust ports and retain the lantern rings in place at the end of the engine stroke. The piston is carried by a piston rod provided at its lower end with a crosshead sliding on two guides, one on either side of the frame. The piston jacket water is admitted and discharged from the jacket through the piston rod by means of swing-joint pipes.

The injection air-compressor of the Carels engine is a four-cylinder, three-

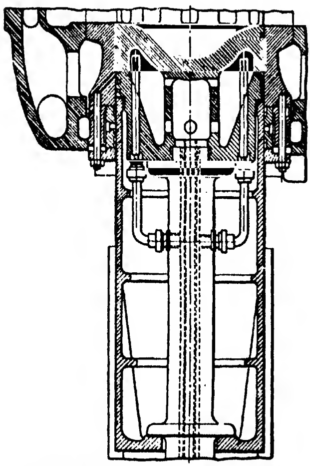


Fig. 103. Section of Carels-Diesel Engine Cylinder
Courtesy of "Engineering", London, England

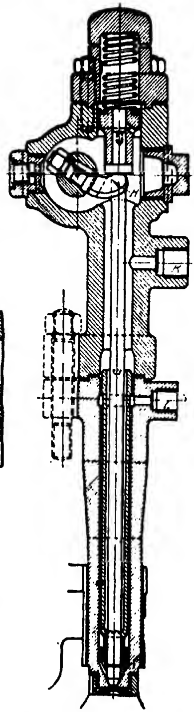


Fig. 104. Fuel Injection Valve of Carels Marine Two-Cycle Diesel Engine
Courtesy of "Engineering", London, England

stage compressor and is driven from the forward end of the main crankshaft by an overhung pin. The scavenging pumps are driven

by links and walking beams from the crossheads of the second and third cylinders, the valves being actuated by eccentrics on the main crankshaft. Air for these pumps is drawn in through silencers to reduce the noise in the engine room. In each elbow, leading from the exhaust chamber around each cylinder to the exhaust header, a sprinkler is located, to extract the heat from the exhaust gases.

The fuel-injection valve is novel in design, in that reciprocating motion through a packed gland is replaced by rotating motion to avoid sticking. This is accomplished by entirely inclosing the injection-valve stem and actuating spindle and packing the oscillating actuating spindle at either end, thus removing the necessity for packing the injection-valve stem. Fig. 104 shows a vertical cross section of the injection valve and atomizer casing. The injection valve is of the pressure-balanced type—balanced by the injection-

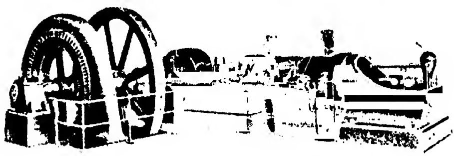


Fig. 105. Beco Oil-Burning Two-Cycle Diesel Engine
Courtesy of The Brown Engine Company, Fitchburg, Massachusetts

air pressure—and thus needs only a light spring to return it to its seat. The fuel-injection valve lever rotates the spindle, to which is keyed the actuator H which raises the valve and admits the fuel and the injection air to the cylinder. The injection-air inlet is at K, and that for the fuel at L.

Reversing is accomplished by the use of two sets of cams—ahead and astern—for each of the fuel-injection and starting-air valve levers and by rotating the camshaft through 32 degrees with respect to its angular relationship to the crankshaft—thereby reversing the opening of the scavenging air valves.

The Diesel motor is coming into use as the auxiliary for sailing ships of the largest size as well as for straight power ships. The leading dimensions of the motor ships launched during 1913 are given in Table XIII.

Beco-Diesel Engine. The engine shown in Fig. 105 is a single-

TABLE XIII

Diesel-Motor Ships of 1913

Name of Ship	Main Engines		Engine Cycle	Cyls. per Engine	Diam. Inches	Stroke Inches	R. P. M.	Gross Tonnage	Type of Vessel	Pro-pellers	Speed Knots
	No.	b. h. p.									
Suecia	2	1650	4	8	$19\frac{11}{16}$	$25\frac{43}{64}$	140	3730	Cargo	2	$10\frac{3}{4}$
Emanuel Nobel	2	2200	4	6	$21\frac{3}{64}$	$39\frac{3}{8}$	125	....	Oil tankship	2	11
Fordonian	1	750	2	4	$18\frac{1}{8}$	$32\frac{1}{4}$	140	2368	Grain carrier,	1	9
Siam	2	2550	4	8	$23\frac{13}{64}$	$31\frac{1}{2}$	125	5295	Cargo	2	$11\frac{1}{4}$
Annam	2	2550	4	8	$23\frac{13}{64}$	$31\frac{1}{2}$	125	5295	Cargo	2	$11\frac{1}{4}$
Pedro Christoffersen	2	1650	4	8	$19\frac{11}{16}$	$25\frac{43}{64}$	140	3730	Cargo	2	$10\frac{3}{4}$
California	2	2300	4	8	$21\frac{17}{64}$	$28\frac{1}{4}$	140	4598	Cargo and passengers	2	$11\frac{1}{4}$
Wotan	1	1650	2	6	$23\frac{5}{8}$	$43\frac{5}{16}$	90	5703	Oil tankship	1	$9\frac{1}{2}$
Loki	2	2100	2	6	$18\frac{45}{64}$	$31\frac{1}{2}$	137	5456	Oil tankship	2	11
Loudon	1	1100	4	6	$22\frac{3}{16}$	$39\frac{3}{8}$	120	1873	Cargo and passengers	1	$10\frac{1}{2}$
Fionia	2	3100	4	6	$29\frac{3}{64}$	$43\frac{5}{16}$	100	5219	Cargo and passengers	2	$13\frac{1}{2}$
Tynemount	2	600	4	6	12	$13\frac{1}{2}$	400	....	Grain carrier	1	9
France	2	900	2	4	$17\frac{23}{32}$	$21\frac{3}{64}$	230	....	Cargo and passengers	2	10
Hagen	2	2100	2	6	$18\frac{45}{64}$	$31\frac{1}{2}$	137	5456	Oil tankship	2	11

cylinder, horizontal, double-acting, two-cycle type of engine—the only valves being the fuel-injection and air-starting valves at each end of the cylinder. These valves are operated by the lay shaft which

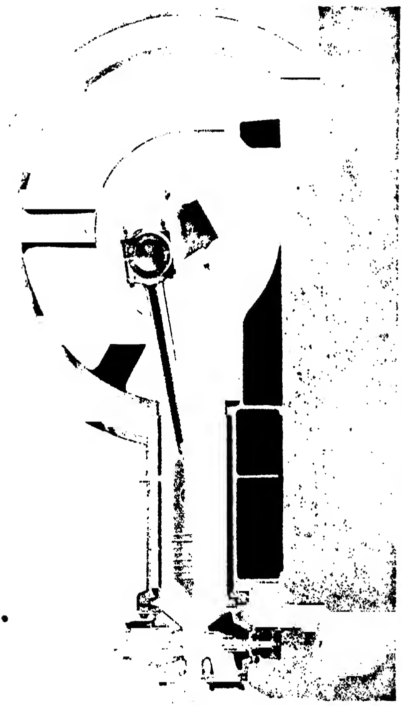


Fig. 106. Longitudinal Section of De La Vergne Oil Engine, Type "FH"
Courtesy of De La Vergne Machine Company, New York City

runs alongside the engine, supported in bracket bearings. The water-cooled piston is supported by a main- and tail-guide crosshead so that it does not wear on the bottom of the cylinder. To the tail-guide

head is attached a vertical walking beam which drives the scavenging and injection-air compressors, located usually on the floor below the engine. The air charge is admitted and the exhaust gases driven out through ports in the cylinder liner, which are uncovered by the piston in exactly the same manner as is the case in a two-port two-cycle gasoline engine.

De La Vergne Modified Diesel Engine. The engine, Figs. 106 and 107, is a combination of the Diesel principle with the hot bulb arrangement. The engine operates on the four-stroke cycle and

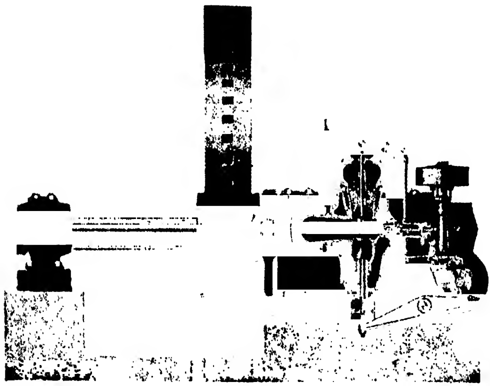


Fig. 107. Transverse Section of De La Vergne Oil Engine, Type "FH"
Courtesy of De La Vergne Machine Company, New York City

compresses the air to 250 to 300 pounds instead of 500 pounds as in the Diesel engine. The temperature thus obtained would not be high enough to ignite the fuel; recourse is therefore taken to a hot bulb, the air in which, owing to the heat radiated from its uncooled walls, attains a higher temperature than that contained in the combustion chamber. At the end of the compression stroke the fuel is injected by means of an air blast of about 600 pounds pressure from the injection nozzle—shown in Fig. 41 and described on page 89—across the combustion chamber intó the hot bulb, where it is immedi-

ately ignited. Owing to the comparatively long distance the fuel spray has to travel after it leaves the nozzle until it is ignited in the hot bulb, ignition first produces a considerable pressure increase (combustion at constant volume), which is followed by combustion at approximately constant pressure. This is shown on the two indicator cards, Fig. 108, where it will be noticed that the maximum pressures are very nearly 500 pounds, i.e., about the same as in the Diesel engine. As far as strains in the engine are concerned there is, therefore, not much difference between these two types. It must be remembered, however, that in the Diesel engine this high pressure must be obtained at the end of the compression stroke in order to secure ignition, while in the De La Vergne engine, ignition is certain at about half that pressure.

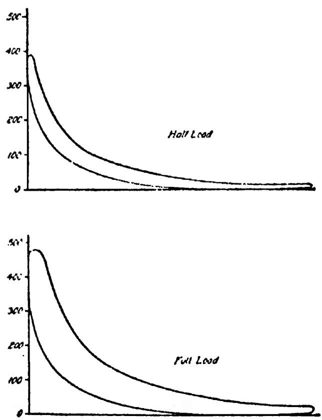


Fig. 108. Typical Indicator Cards Taken from Type "FH" Engine
Courtesy of De La Vergne Machine Company, New York City

The details of mechanical design are very similar to Type "HA" engine shown in Figs. 91, 92, and 93 and described on pages 153 to 155. The vaporizers in the two engines may be similar, the only difference being that in Type "FH" it is located in the side of the cylinder head opposite the injection valve instead of in the end of the cylinder head. The inlet and exhaust valves are located in the cylinder head, one over the other, and are actuated by cams and levers. The injection-air system on this engine is that described on page 174, the high-pressure stage of the compressor handling only the amount of air necessary at that moment for injection. The two-stage compressor is mounted on the side of the engine frame and is driven by an eccentric on the crankshaft.

The governor, on this type of engine, besides regulating the amount of the air charge admitted to the high-pressure stage of the compressor, also regulates the amount of oil injected by varying the stroke of the oil pump to suit load conditions. The method of obtaining this effect is shown in the transverse section, Fig. 107. An eccentric, mounted on the end of the lay shaft, has a link pinned at one end to the eccentric strap and pivoted to the bracket bearing at the other. Between this link and the bell-crank lever, the long arm of which is parallel to the link, is a roller, the position of which is regulated by the governor. As the engine speeds up, the governor rises, pulls the roller up with it, and reduces the amount of motion transmitted through the roller by the link to the bell-crank lever, until, when the governor is at the top of its travel, no motion is transmitted. The short arm of the bell-crank lever is attached to the oil-pump plunger and forces it on its down stroke, the return stroke being made under the action of a coil spring acting on the plunger.

DIESEL FUEL-OIL PUMPS

Characteristics of Pump System. In the Diesel engines with closed-nozzle injection, the space into which the oil is delivered is always in communication with the injection air at a pressure of from 750 to 1000 pounds, so that the oil must be delivered against this high pressure. The quantity of oil to be delivered is small, especially in the case of the original Diesel engine where there was a separate pump for each cylinder. Since the work of the fuel-oil pump is in no direct relation to the cycle of events in the cylinder, it is possible to use but one pump for all the cylinders in a multicylinder engine. The pump delivers the oil into a distributor where, by means of a series of check valves and restricted passages which artificially increase the resistance to flow, it is equally divided into as many streams as there are cylinders.

Type of Plunger. Positively operated plungers (operated by an eccentric) are generally preferred to those operated by a cam. In the latter, the stroke is varied according to the load by letting the governor shift a wedge block in between the cam and the plunger. In the former, the displacement of the plunger is constant and larger than that required for the maximum charge of fuel oil; the excess oil is discharged through the suction valve, the opening and closing of

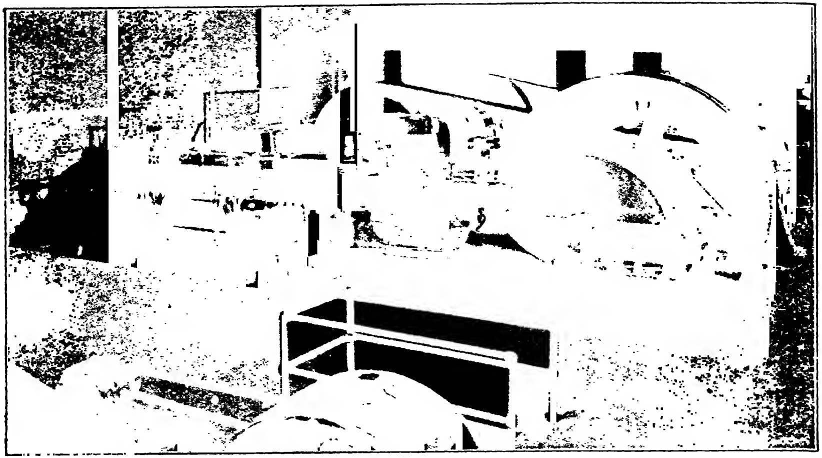
which is determined by the position of the governor. This valve and also the mechanism for its operation are constructive elements which require the utmost care in design as well as workmanship, as the accuracy of fuel-oil delivery depends, primarily, upon their proper action. Frictional resistance of the valve, which is likely to prevent its prompt closing, is especially important and must be reduced to a minimum. It is therefore not advisable to have the valve stem pass through a stuffing box. The best practice at present is to use a positively operated plunger for the operation of the inlet valve, this plunger passing through the stuffing box and thus practically eliminating friction as far as the valve is concerned.

In the case of the open nozzle, the fact that the oil is delivered at the start of the compression stroke against the low pressure then existing in the working cylinder makes the oil pump a comparatively simple piece of apparatus.

INJECTION AIR SUPPLY

Use of Two- or Three-Stage Compressor. In all the various types of modern oil engines the apparatus required to obtain the high-pressure injection air forms a comparatively complicated, and therefore expensive, accessory. A two- or three-stage compressor must be provided, in which the suction valve, on either the low- or high-pressure stage, must be fitted with a governor-regulated adjusting device to vary the amount of air drawn in, according to the oil charge. Intercoolers between the stages and an after-cooler after the high-pressure stage must be provided to keep the air cool enough to prevent pre-ignition in the injection nozzle.

Storage Tanks. In most engines there is provided a tank or tanks for the storage of the injection air, and this tank must be fitted with three valves; one to close it off toward the engine and one toward the compressor, and one safety valve. In one of the modified Diesel engines, the low-pressure stage of the compressor discharges into storage tanks from which the high-pressure stage draws its air, the amount being regulated by a governor-operated valve to suit the varying charges of oil at each injection. The high-pressure air is discharged directly into the injection-valve cage, without the use of an intermediary storage tank. This system has the advantage that the air is stored at low pressure, from 125 to 150



MUENZEL SUCTION PRODUCER GAS PLANT WITH 400-HORSEPOWER MUENZEL DOUBLE-ACTING TANDEM ENGINE IN FOREGROUND

Courtesy of Minneapolis Steel and Machinery Company, Minneapolis, Minnesota

pounds instead of 750 to 1000 pounds, which results in lighter tanks that, with the connecting piping, are more easily kept tight. On the other hand, it is necessary that the high-pressure stage of the compressor should be reliable.

IGNITION SYSTEMS

Ignition Requirements. For satisfactory action of the engine, the ignition of the explosive mixture must be certain, and must occur at a definite, predetermined time. In timing the ignition, it has to be recognized that the explosion is not instantaneous, but requires a not inconsiderable period of time to arrive at the maximum pressure. The actual duration of the explosion depends on the strength of the explosive mixture and on the amount of compression to which it is subjected. The ignition should have *lead*—that is, should begin before the end of the return or compression stroke, so that the maximum pressure is reached when the crank has just passed the dead center. The amount of lead varies with the speed, strength of mixture, and other conditions. The indicator card *a*, Fig. 109, is the correct diagram with properly timed ignition. If the ignition is later than this, indicator cards similar to *b* or *c* will be obtained, and the engine will do less work and be less efficient. If the ignition is too early, the maximum pressure will be obtained, Fig. 110, before the crank has reached its dead center, and will tend to reverse the engine. This causes great shock to the engine, its rapid deterioration, and lowered efficiency. The immediate external evidence of too

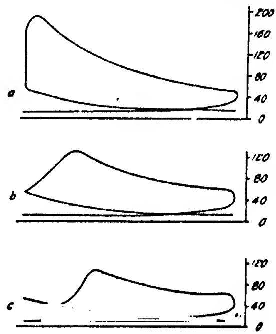


Fig. 109. Indicator Cards with Variously Timed Ignition

Fig. 110. Indicator Card with Too Early Ignition

the engine. This causes great shock to the engine, its rapid deterioration, and lowered efficiency. The immediate external evidence of too early ignition is a violent pounding noise in the engine. Two methods of ignition are in common use in engines using the Otto cycle. The first is by exploding the mixture by contact with a surface hot enough to cause ignition; the second is by means of an electric arc.

HOT-TUBE IGNITION

Method of Operation. A hot tube was the common device when the first method of ignition was used; this method has fallen into disuse in this country with improvement in electric ignition; in England, however, this method is still used occasionally on engines using illuminating gas. The tube E, Fig. 111, is closed at the upper end, and communicates at its lower end through the port B with the cylinder A. It is heated by an external flame from the Bunsen burner C, and is maintained at a full red heat. The chimney around the tube is lined with asbestos, and keeps the flame in good contact with the tube. During the admission stroke the tube is filled with products of combustion at atmospheric pressure remaining from the previous explosion. As compression goes on, the nonexplosive products of combustion are crowded into the upper part of the tube, while part of the explosive mixture in the cylinder is compressed into the lower part of the tube. The length of the tube and the position of the flame are adjusted by experiment, so that the explosive charge will just reach the hot portion of the tube and be ignited at the moment when ignition is desired. Shortening the tube makes the ignition come later. With this device the actual time of ignition is not

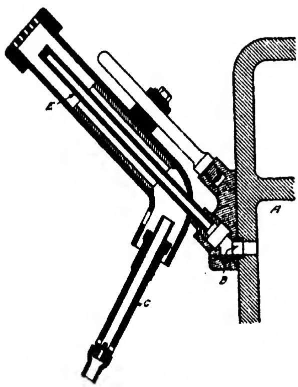


Fig. 111. Hot-Tube Igniter

very definite. It depends on the temperature of the tube, the position of the Bunsen flame, the strength of the mixture, and the amount of compression. As these last two quantities are purposely varied by the governor in some engines, irregular timing would result from its use in such cases.

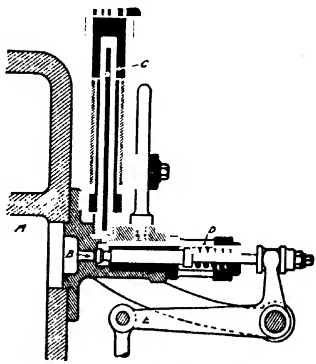


Fig. 112. Hot-Tube Igniter with Timing Valve

Use of Timing Valve. The irregularity of timing with the hot-tube igniter can be partly remedied by the use of a timing valve. The timing valve *B*, Fig. 112, is held on its seat by a spiral spring *D* until ignition is desired, when, by a movement of the bell-crank lever *E*, the valve opens and the compressed charge in the cylinder *A* gets access to the hot tube *C*. The valve *B* is kept open until the end of the exhaust stroke. The tubes are preferably made of nickel alloy or of porcelain, but the latter is very brittle and apt to break when being fastened in place. Iron tubes are used sometimes, but they burn out rapidly and are unreliable.

Fig. 113. Spark Coil
Courtesy of Thordarson Electric Company, Chicago

ELECTRIC IGNITION

Even when provided with a timing valve, the hot tube does not give very satisfactory ignition; and, moreover, some time is consumed in heating the tube before the engine can be started. Accurate timing can be obtained best by electric means, and electric ignition is consequently used more than any other.

Method. The method is to make a spark pass, at the instant when ignition is desired, between two terminals situated in the clearance space of the engine. The most common way of forming the spark is to separate two contact points through which a

current has been flowing. An electric arc will then pass between the separating contact points. In order to insure that the temperature of the arc is high enough and its duration sufficient to ignite the explosive mixture through which it passes, a spark coil, or choking coil, is generally inserted in the circuit. A spark coil, Fig. 113, consists merely of a bundle of soft-iron wires, surrounded by a coil of insulated copper wire through which the current passes. The contact points of the igniter must be brought together to re-establish the current before another spark can be obtained. A device of this nature is known as a *make-and-break igniter;* and when the contact points do not slide across each other, it is called a *hammer-break contact.*

Behavior of Current When Contact Is Broken. The action of the electric current when the circuit is broken is analogous to a stream of water whose flow is abruptly dammed. The flow cannot be stopped instantaneously, since the stream has a certain amount of kinetic energy due to the weight of the water flowing and the velocity of flow, and this causes it to attempt to overcome the stoppage. This kinetic energy cannot disappear and be lost and is, therefore, changed to pressure, which tends to blow out the dam. If this stoppage occurs in a very long pipe line the weight of water flowing will be greater than in a shorter line, in proportion to the relative lengths of the lines; therefore, the flow of a greater weight of water must be stopped and. consequently, the resulting pressure against the dam will be higher.

When the electric current is dammed by the contact being broken, it also tries to keep on flowing, the pressure builds up, and if the volume and length of stream flowing is sufficient to build up pressure enough, the current jumps, or arcs, across the gap. If the length of current flowing is increased by putting a choking coil in the circuit, the pressure resulting from the break is increased, as in the case of the longer water line. and will insure the current arcing across the gap, thus making the ignition certain. The choking coils usually used for ignition purposes produce an instantaneous pressure in the circuit, following the break, of about 5000 volts.

Make-and-Break Ignition

Ordinary Types. One of the common forms of hammer-break igniter is illustrated in Fig. 114, which shows an igniter plug removed from the cylinder head. The movable electrode *b* is at the end of an

arm fastened to the spindle *c*. When the interrupter lever *d*, which is loose on the spindle *c* and is connected to it through a coil spring, is lifted by an arm from the camshaft of the engine, it rotates the spindle *c* so as to bring *b* into hard contact with the stationary and thoroughly insulated electrode *a*. This completes a circuit and permits a current to flow from *a* to *b*. When ignition is desired, the lever *d* is tripped and flies back, carrying with it the spindle *c*, abruptly breaking the contact and causing an electric arc to form between *a* and *b*. The contact points are generally made of platinum, as this does not oxidize or corrode; but other metals are also used. The passage of the spark takes minute particles of the material from one terminal and deposits them on the other,

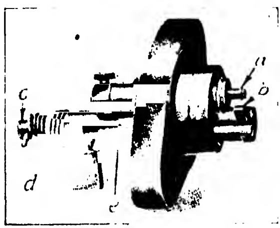


Fig. 114. Hammer-Break Igniter Plug Removed from Cylinder Head
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

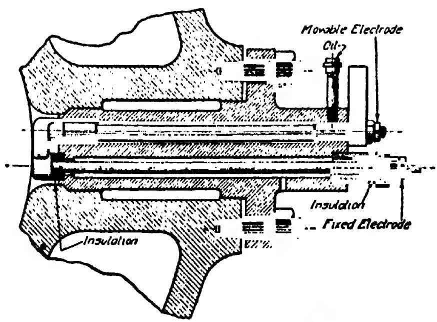


Fig. 115. Igniter Plug of Crossley Gas Engine

the action following the direction of the current. By reversing the current, the material is returned to the terminal from which it was taken, thus increasing the durability of the contact points.

An English form of igniter plug is shown in Fig. 115.

Wipe Break. A make-and-break contact is sometimes obtained by sliding one contact point over the other until it slides off completely. This is known as a "wipe" break. The method insures a good contact, produces a very hot spark, keeps the contact points clean, but wears them out quite rapidly. Provision must be made for adjustment, otherwise the timing will alter with the wear of the points. The rubbing surfaces can be of steel.

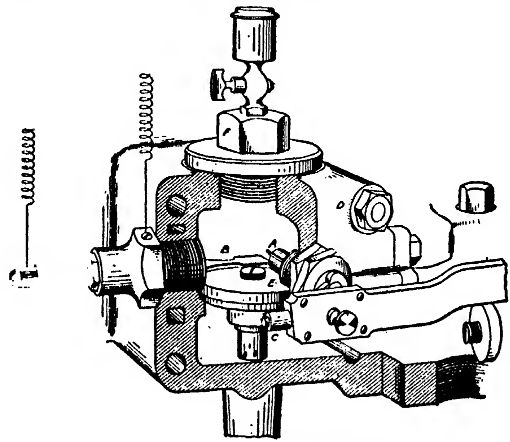


Fig. 116. Foos Revolving Wipe-Contact Electric Igniter

An example of the wipe-contact igniter is shown in Fig. 116. The stationary electrode *B* is a flat steel spring; the moving electrode *A*, which is rotated by the igniter rod, comes in contact with *B* once per revolution, thus establishing the circuit, presses *B* down, and finally trips it. The abrupt breaking of the circuit causes a good spark to pass between the electrodes. The igniter is placed immediately over the inlet valve *E*. The thumbscrew *C* on the igniter rod permits the adjustment of the time of the ignition.

Hammer Break. The igniter gear of an engine with hammer-break ignition is shown in Fig. 117. The igniter rod *f*, which is sup-

ported on the reel *h*, receives a reciprocating motion from a crank *g* at the end of the side shaft. During the exhaust or admission stroke, the end of the rod *f* comes in contact with the interrupter lever *d*, as may be seen by comparison with Fig. 114, and establishes the contact of the electrodes. The vertical component of the movement of the end of the rod *f* sets free the lever *d* at the moment when ignition is desired.

The make-and-break igniters so far shown are used principally on small and medium-power engines. The electrical contact is generally made between pins carried in the stationary and movable electrodes. These pins, if the engine is small enough, are made of platinum. Platinum is expensive to replace, especially if the engine is large, thus calling for large points and, therefore, steel pins are used, which give very nearly as good ignition as platinum points.

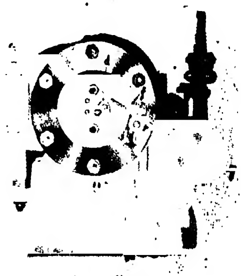


Fig. 117. Hammer-Break Igniter Gear of Root and Van Dervoort Engine

Igniters for Large Engines. *Ordinary Type.* The design of igniter plugs for large engines differs only in details from that shown in Fig. 114. For instance, instead of insulating only the stationary electrode, both the stationary and movable electrodes are thoroughly insulated in order to reduce to a minimum the chances for a shutdown due to a short circuit. With only one electrode insulated, if that becomes grounded or short-circuited, the igniter is out of commission; whereas, with both electrodes insulated, they must both be grounded before the igniter will be put out of use. Another point of difference between igniters for large and small engines is that for large engines the electrodes have no points or pins. The movable electrode head, or hammer, is made of cast iron and the stationary electrode, or anvil, of low-carbon steel. With these electrodes the oxidization caused by the electric arc together with that caused by the heat of combustion are small and, therefore, the igniters can be run for long periods without cleaning of the contacts.

With the make-and-break igniters so far described, it is necessary to have a separate tripping device driven from the lay shaft for each igniter. In a large double-acting engine, with two igniters at each end of the cylinder, this means that four separate tripping devices are required on each cylinder, resulting in complication and considerable noise.

Magnetic Make-and-Break Type. It is possible, practically, to eliminate the complication and the noise by substituting for the mechanical make-and-break device an electrically actuated make-and-break apparatus. Such a piece of apparatus is called a *magnetic plug;* it must always be used in series with a distributor or timer which sends current to it at the instant when a spark is desired.

In Fig. 118 are shown, to the left, the electromagnetic device, and to the right, the outside, of a make-and-break plug similar to that shown in Fig. 114. On passing current through the electromagnet, the armature, which is one arm of a bell-crank lever, is attracted to the magnet, and the other arm of the bell-crank lever strikes the moving electrode. The electromagnet is used in series with one of the electrodes. The circuit is re-established by the action of gravity.

Fig. 118 Magnetic Make-and-Break Igniter
Courtesy of Westinghouse Machine Company, Pittsburgh, Pennsylvania

A diagram of the wiring from any suitable source of electricity to four magnetic igniters is given in Fig. 119. The simultaneous adjustment of the timing of the four igniters is effected by rotating the timer through the desired angle.

While these magnetic plugs reduce the noise and the complication, they have the bad feature that it is almost an impossibility to time the ignition accurately because of the effect of magnetic lag. While their use for large engines was almost universal a few years ago, they are now used by only a few engine builders, in fact, some of the engines that were equipped with magnetic plugs when first built have been re-equipped with mechanical make-and-break plugs.

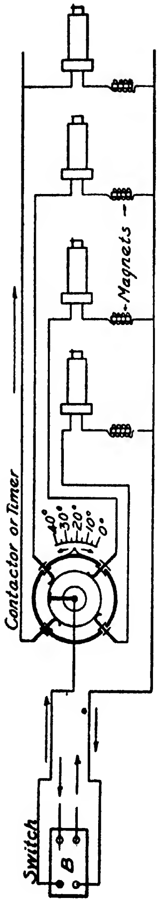


Fig. 119. Wiring Diagram for Magnetic Igniters

Mechanical Make-and-Break Type: The igniter shown in Fig. 120 is of the most advanced design. The tripping device is actuated by an igniter camshaft *A*, which runs throughout the length of the cylinders and is driven from the lay shaft through a ratchet, so that if the engine shaft oscillates or reverses in starting or stopping, the igniter camshaft will not reverse and wreck the tripping mechanism. The cam *B* is a knock-off cam, the cam gradually increases in diameter and then abruptly drops to the original diameter so that the push rod *C* is gradually raised and then quickly returned to its original position. The raising of the push rod is resisted by the spring *D*, which rests at one end in the bracket carrying the push rod and at the other on a collar on the push rod. The top of the bracket is fitted with a rubber bumper *E*. The collar *F* on the push rod is so adjusted that the push rod does not strike the small part of the cam when the knock-off occurs—in this way there is no click of metal on metal, and the rubber bumper can do its full cushioning work. Above this collar is a hard-rubber disk *G*, mounted on a threaded steel sleeve, so that its position on the push rod may be adjusted. The disk *G* serves as the hammer, which abruptly moves the movable electrode *H* and breaks the circuit. The spring is attached to *H*, which tends to hold the electrodes in contact. When the push rod is at the highest point of its travel, just before the knock-off, the disk *G* is set with a clearance of an eighth of an inch between it and the movable electrode arm. When the knock-off occurs, the push rod has

some distance in which to travel and gain velocity before striking the movable electrode arm, thus causing an abrupt break of the circuit, and in addition this clearance insures a contact between the electrodes before the break is made. The push-rod bracket is hinged and held in place by a bolt. The push rod can be made to knock off sooner or later by moving the bracket out or in, and thus the timing of the individual igniters can be changed while the engine is running. The bracket

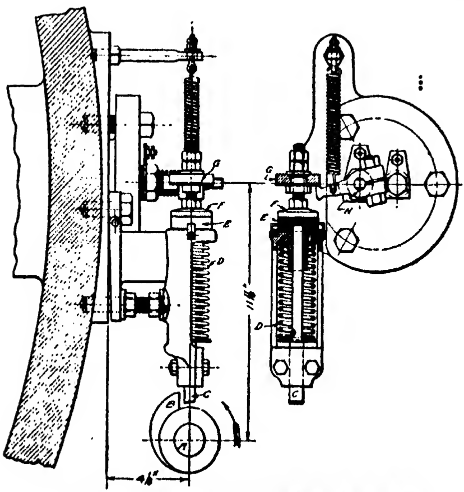


Fig. 120. Mechanically Operated Igniter for Allis-Chalmers Gas Engine
Courtesy of Allis-Chalmers Company, Milwaukee, Wisconsin

is bolted to a pad on the cylinder, clear of the igniter plug, so that the latter can be removed for inspection and cleaning without disturbing the bracket. The stationary electrode is a steel rod on which the anvil is machined. The movable electrode consists of a steel rod screwed into a cast-iron head or hammer, and the end of the screw riveted over. It is mounted in a steel tube, a ground 45-degree joint being provided where the head seats in the tube to prevent leakage from the cylinder out through the igniter. The stationary electrode

and the tube containing the movable electrode are insulated from the plug at both ends by mica washers, care being taken that the holes through the plug are large enough to prevent either touching the plug and grounding. A light spring is provided under the arm of the movable electrode, just stiff enough to prevent the unseating of the head during the suction stroke.

Ignition Current

Batteries. In small engines the current is commonly taken from a primary battery, consisting of about five cells. The Edison-Lalande cell, made up of two zinc plates and a plate of compressed copper oxide immersed in a strong solution of caustic soda, is perhaps the most largely used. Dry cells and storage-battery cells are also used. Current is sometimes taken from a direct-current lighting or power circuit; but this is objectionable, because the circuit is grounded every time the igniter terminals are in contact.

Dynamo or Motor-Generator Set. The practice is growing, of using either a small special dynamo or a magneto-dynamo for the

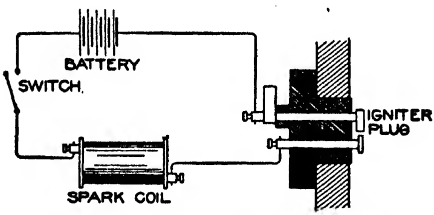


Fig. 121. Diagram of Circuit for Make-and-Break Ignition

exclusive purpose of supplying the current for ignition. This makes the ignition spark more certain and of more uniform strength than when a battery is used, as the latter deteriorates with use. When a generator is used to supply current to a make-and-break igniter, lamps—one or two in parallel—are used in series with the igniter to govern the amount of current; otherwise, too much current would flow and the contacts of the igniter would be rapidly burned away.

A switch, Fig. 121, should always be included in the electric circuit, and should be thrown out when the engine is not running,

in order to prevent the short-circuiting and consequent exhaustion of the batteries.

When multicylinder engines are used, the ignition circuit is closed for a large proportion of the total running time, and consequently the batteries will run down rapidly. To eliminate the trouble and annoyance of frequent refilling of the primary cells, or of frequent recharging of the storage batteries, it has become usual in large engines, and very common in small engines, to generate the current required for ignition by mechanical power. The simplest means of accomplishing this is by the use of a small dynamo driven by the engine; but this generates a current whose amount depends on the speed of the engine. A battery must be employed to start the engine; when the speed of the dynamo is sufficient to give the desired current, the battery is thrown out of the ignition circuit, and the dynamo is put in by means of a double-throw switch. As the ordinary ignition dynamos have self-excited field magnets, the current generated increases in a double ratio with increase of speed; that is, not only is the armature speed increased, but the field excitation is increased also. The result is that, as the speed increases, a dynamo is likely to give an excessively hot spark, which tends to burn away the contact points rapidly. Consequently, a dynamo is best used on a constant-speed engine. If used on a variable-speed engine, it is necessary to have a *speed governor*, which prevents the generator acquiring more than a certain desired speed; without this, the current at high speed might be destructive to the generator. The electrical output of a dynamo is large compared with that of a magneto of the same size and speed.

In large gas-engine-driven electric-generating stations, the ignition current is supplied by a motor-generator set, which is used for no other purpose. The engines are started on a storage battery and when the main generators are excited, the ignition motor-generator set is started and run on the power generated by the engines and the ignition switched from the battery to the motor-generator circuit. In some plants, where an outside source of current to run the motor-generator set is available, the storage battery is dispensed with.

A dynamo may be used either with make-and-break ignition or with jump-spark ignition. In the former case, it is not absolutely necessary to have a spark coil in series, as the self-induction of the arma-

ture furnishes the necessary extra current when the circuit is broken; a coil is generally used, however, to increase the pressure after the break. With jump-spark ignition, the usual induction coil is necessary.

Magnetos. *General Characteristics.* With a variable-speed engine, if a mechanically generated current is to be used, it is best obtained from a magneto. The only fundamental difference between a magneto and a dynamo is that a magneto has permanent magnets, while a dynamo has electromagnets. The strength of the magnetic field through which the armature rotates will naturally remain constant in a magneto, while with a self-excited dynamo it increases with the speed. The variation of the current generated with the speed, will consequently be less in a magneto than in a dynamo. A magneto may be run in either direction. For ignition purposes it is not necessary that the current should have constant direction; consequently,

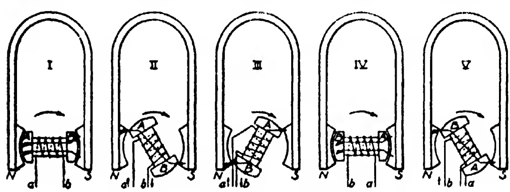


Fig. 122. Position of Rotating Armature of Magneto

ignition magnetos are not supplied with the usual commutators and brushes, and they deliver an alternating current. One terminal of the armature coil is grounded on the armature core; the other goes to a collector ring on the shaft, and is taken off by a single brush.

The magneto armature may be constructed precisely like a dynamo armature with commutators and brushes, delivering continuous current. In that case it does not have to run in step with the engine, but requires a spark coil.

More frequently the armature is of the *H* type, Fig. 122, with a single coil of comparatively coarse wire. The motion of the armature may be either a continuous rotation or an oscillation. With continuous rotation, the current induced in the armature goes from zero to a maximum, twice in every revolution. For the spark, the circuit is preferably broken when the armature current has its maxi-

mum value. This is readily accomplished in magnetos which are geared directly to the engine by making the speed of the magneto the proper multiple of the speed of the engine, the position of maximum voltage of the magneto being made to coincide with the explosion position of the engine. When the magneto and engine have the desired relative speeds and positions, they are said to be *running in step* or *in synchronism*.

Since the time of ignition of an engine should be made earlier as the speed becomes greater, it is desirable that the relative positions of magneto and engine should be capable of slight adjustment while the engine is running. This is accomplished in various ways. It is not, however, always necessary to break the circuit at the point where the current is greatest, since there is considerable current flowing, as seen in Fig. 123, for some time after the magneto has passed its position of maximum current. Consequently, if the relative positions of magneto and engine are fixed once for all, so that the current is at its maximum when the circuit is broken at the highest speed, then, when the circuit is broken with a smaller advance at some lower speed, there will be sufficient current for a satisfactory spark. Owing to the intensity of the magneto current, the advance of the spark that is required in order to produce a satisfactory explosion is considerably less than is necessary in the case of a battery current. It is, of course, most necessary to have a hot spark when the speed is highest.

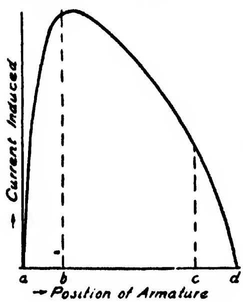


Fig. 123. Curve Showing Current Induced by Magneto of Figs. 122 and 124

The voltage of a magneto naturally increases with speed; but the rate of the increase is not nearly so great as that of the speed, on account of the reaction of the armature on the comparatively weak permanent field.

Magnetos are used either for make-and-break or for jump-spark ignition. In the former case, they are low-tension magnetos; in the latter, high-tension magnetos.

Electrical Action. Magnetos are of two types: (1) those with *rotating armature;* and (2) the *inductor type,* with stationary armature and rotating segments or inductors. In both types—as in all electromagnetic machinery—the generation of electromotive force results from changes in the number of interlinkages between magnetic lines of force and the coils of an electric conducting circuit. The number of interlinkages which any one line of force makes with a closed coil of wire, is the number of turns or loops that it traverses. The interlinkage in any electromagnetic apparatus is the sum of the interlinkages of all the magnetic lines of force.

The voltage (and current) induced at any instant is proportionate to the rate of change of interlinkage. Consequently, no current is generated when the interlinkage is a maximum, for at that time the rate of change of interlinkage is zero.

The *direction of flow* of the induced current depends on two things: (1) Whether the interlinkage is decreasing or increasing; and (2) the direction in which the magnetic lines of force thread the coil. A decrease in the interlinkage with the lines of force threading the coil in one direction, gives a current in the same direction as an increase in the interlinkage when the direction of the lines of force is reversed.

Rotating Armature Type. In Fig. 122, several positions are shown of the rotating H-shaped armature. The long arrows passing through the armature represent the lines of force passing from the N pole to the S pole. The winding ab of the armature is shown diagrammatically, and the direction of flow of the current induced in it is indicated by the arrows. In position I, all the magnetic lines of force pass through the armature; the flux is consequently a maximum, and the induced current zero. As the armature rotates to position II, the number of lines of force actually threading the armature decreases, and a current is induced in the direction shown. From position II, the magnetic flux continues to decrease till it becomes zero, when the armature is in the vertical position. From there on, the magnetic flux in the armature reverses itself; that is, instead of going from A to B, it goes from B to A, and increases in amount as the armature rotates through position III to position IV, where it again reaches its maximum value. The effect of the increase of the reversed flux is to give an induced e.m.f. in the same direction as that resulting

from the decrease of flux in the original direction. Consequently, the e.m.f. is in the same direction while the armature rotates from position I to position IV; it starts and ends at zero, and has its maximum value while the rate of change of flux is greatest—that is, between positions II and III. During the other half of the revolution, while the armature is rotating clockwise from position IV to position I, a similar action takes place; but the e.m.f. is all the time in the reversed direction, as indicated in position V. The current lags somewhat behind the e.m.f.

The variations in the duration or magnitude of the induced current depend principally on the design of the pole pieces and armature. The magneto shown in Fig. 124 gives the induced current represented in Fig. 123 while it is moving between the two positions shown—that is, while it is moving through an angle of 25°. The result of one complete revolution of the magneto is an induced current for about 25° of rotation; very little current for the next 155°; a current in the reversed direction for the next 25°; and very little current for the remaining 155°. This magneto may then be used for ignition twice in its revolution. If it is used with (1) a single-acting four-cylinder four-cycle engine; or (2) a double-acting two-cylinder four-cycle engine; or (3) a single-acting two-cylinder two-cycle engine; or (4) a double-acting one-cylinder two-cycle engine, the magneto should run at the same speed as the engine. With a single-acting six-cylinder four-cycle engine, it must run at one and one-half times the engine speed, in order to ignite all six cylinders.

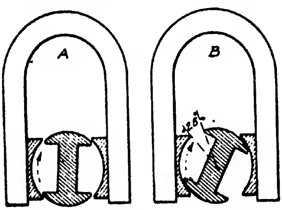


Fig. 124. Magneto Giving Induced Current During Part of Rotation of Armature

The ignition can occur only during the comparatively short period while current is being induced, and should occur preferably when the induced current is at or near its maximum. The form of the current curve, Fig. 123, is then of importance in determining the permissible variation in the point of ignition while the magneto and engine keep in step. A magneto with a current curve that keeps up

well, may permit as much variation in the ignition as is desired. In Fig. 123, for example, the current will be ample for ignition from *b* to *c* (that is, for a rotation of the armature of about 15 degrees), so that there is the possibility of changing the ignition through a range of 15°, if the engine and magneto run at the same speed.

A variation of 15° between the earliest and latest possible ignition will be ample for some engines, but may be insufficient for others. Variation in the ignition is employed when starting up an engine; and also for variable-speed motors, such as automobile engines, when the ignition has to be made earlier as the speed increases, so as to give

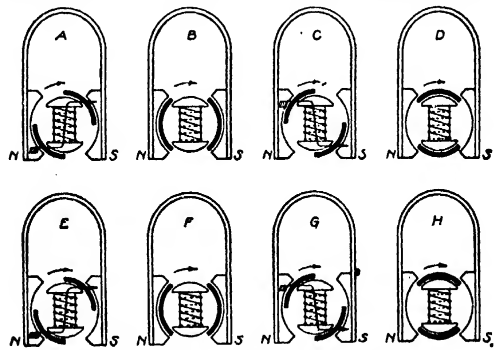


Fig. 125. Diagrams Illustrating Action of Inductor Magneto

time enough for the combustion to be fairly complete shortly after the beginning of the stroke.

If the design of the magneto is such that the current curve has a sharp peak—that is, the duration of a current sufficient for ignition is quite short—or if the desired variation of ignition is, for any other reason, greater than the duration of an adequate current in the armature coil, there must be some device for adjusting the relation of the magneto to the engine so as to make the peak of the current curve coincide with any desired point of ignition.

The simplest way of accomplishing this is to rotate the magneto shaft with reference to the engine shaft by the use of a sleeve with

an external spiral groove or an internal straight feather, which is interposed between the armature shaft and its pinion. By a longitudinal movement of this sleeve, the armature is rotated in relation to the engine shaft.

Inductor Type. The action of the inductor magneto is shown in Fig. 125. This figure shows eight positions in the rotation of the inductor, which consists of two cylindrical segments of soft iron. The magnetic condition of the armature core depends entirely upon the position of the inductor, which, in turn, is determined by the engine position, the inductor being geared to, and running synchronously with, the engine. The armature is stationary. In the positions *A*, *C*, *E*, and *G* the segments form a magnetic bridge between the magnet poles and the heads of the armature core, and the core becomes highly magnetized. The path of the magnetic lines is shown in the diagram. In these positions, there is maximum interlinkage. In passing through positions *B*, *D*, *F*, and *H*, the magnetic lines are abruptly changed in their direction, and a vigorous induced current is set up, both from the breaking down of the existing lines of force and from the setting up of new lines in the opposed direction. This reversal occurs four times during one revolution of the inductor; and succeeding reversals give current in opposite directions. Consequently, the inductor magneto gives twice as many electrical impulses per revolution, and, consequently, has to be rotated only half as fast, as the rotating-armature type of magneto. Since the winding is all stationary, no brush is needed to take the current from

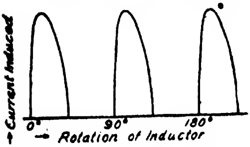


Fig. 126. Current Curves of an Inductor Magneto

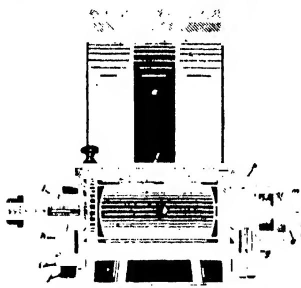


Fig. 127. Bosch Low-Tension Magneto

the armature; all the electrical connections to the armature are stationary. Typical current curves of an inductor magneto are given in Fig. 126.

The construction of a simple magneto is shown in longitudinal section in Fig. 127. The armature *b*, carrying the winding, rotates in the bearings *f* and *g* between the poles of the magnets *a*. One end of the winding is fastened to the armature core; the other end goes to the contact piece *c*, which passes through the hollow armature spindle and is insulated from it. The current is taken from this contact piece *c* by the carbon *e*, which is pressed against it by a spiral spring,

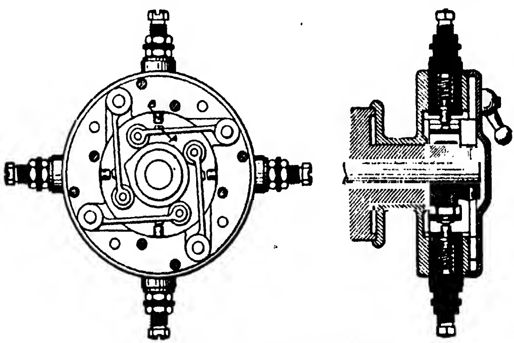


Fig. 128. Side and Sectional View of Pittsfield Timer

and which is insulated by the soapstone disk *m*. The carbon *k*, which is pressed against the body of the armature by a spiral spring, gives a good electrical contact between the rotating armature and the frame of the magneto. Such a magneto is called a *low-tension magneto*. If it is to supply current for a number of igniters, a *timer* or *distributor* must be used with it; this distributor must be geared to the magneto in such way as to insure a sufficient current being generated at the moment when the circuit is established with each of the igniters. The distributor is usually a rotating metal segment connected with the insulated terminal of the armature coil, coming in successive contact with conductors leading to the insulated stationary electrodes of the igniters.

Timers. The timer shown in Fig. 128 consists of a cam (driven from the lay shaft) the high point of which comes in contact once during a revolution with the rollers on each of the four pivoted arms. Each arm has a contact point *A*, which is thereby brought into contact with the insulated and spring-supported contact point *B* connected with each of the four terminals. The contact points are held firmly together by the springs until the cam passes. There are coiled springs in the pivot ends of the arms, which hold the rollers in contact with the cam.

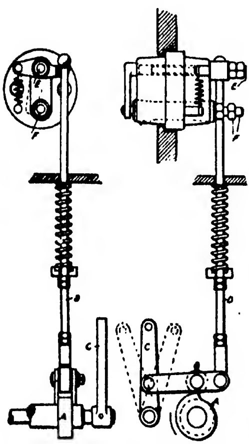


Fig. 129. Bosch Mechanical Make-and Break Igniter

Low-Tension Magneto. The current from a low-tension magneto is used only for make-and-break ignition. An example of its use is given in Fig. 129, which shows a mechanical make-and-break apparatus with a device for varying the time of ignition. The cam *A* on the lay shaft, working through the lever *B* and rod *D*, brings the moving electrode *E* into contact with the stationary and insulated electrode *F*. At a certain position of the cam, this contact is suddenly broken by the action of the spring on the rod *D*. The interruption of the circuit must be made to occur while current is being induced in the magneto. A moderate variation in the time of the ignition is obtainable by shifting the lever *C*, which shifts the position of the roller on the lever *B*; a movement to the right makes the ignition earlier; to the left, later.

Oscillating Magneto. With such magnetos as those already described, if the armature is rotated in synchronism with the engine, its speed will be low when the engine speed is low, and consequently the current will be feeble at that time. It may be necessary, therefore, to have some supplementary source of electricity which can be switched on to the igniter circuit when the engine is being started,

or when, through overload or other cause, it slows down below a certain speed. This auxiliary source may be either a battery or a separately driven generator. It is possible, however, to construct an engine-driven magneto which shall give a current the amount of

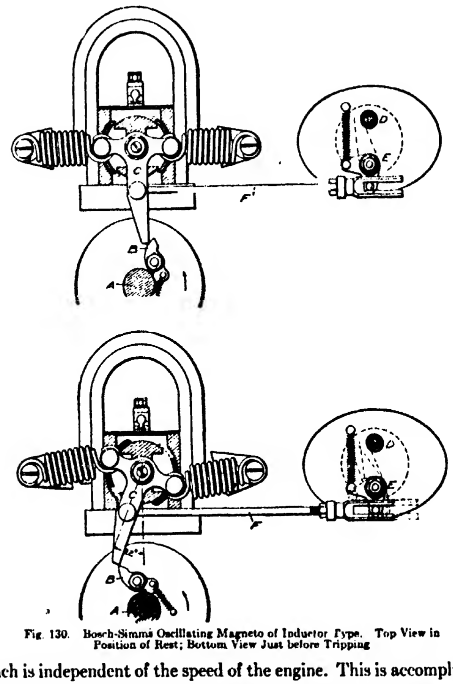


Fig. 130. Bosch-Simms Oscillating Magneto of Inductor Type. Top View in Position of Rest; Bottom View Just before Tripping

which is independent of the speed of the engine. This is accomplished by giving the moving part of the magneto an oscillating instead of a rotary motion, and making the current-generating movement occur at a predetermined time through the action of a stressed spring, and, consequently, at a speed which does not depend on the engine speed. Such an arrangement is often used on stationary engines.

In Fig. 130 is shown an oscillating magneto of the inductor type —first, in its position of rest; and second, in its position immediately before tripping. The moving cylindrical segments are fastened to the T-shaped lever C, whose fulcrum coincides with the axis of the segments. The lower end of this lever is moved through the range shown in the figure, by a lifter B, which is fastened to the lay shaft A. The extremities of the upper arms of the lever are held by strong spiral springs which tend to keep it in the first position. On the rotation of the lay shaft past the second position, the lever is tripped, and the springs bring it smartly back; and, after a few rapid oscillations, it comes to rest again in the first position. This rapid oscillation gives rise to a rapid succession of electrical impulses.

The connection of such a magneto to the make-and-break ignition apparatus, is shown in the same figure. The fixed electrode D is electrically connected to the insulated terminal of the magneto; E is the other electrode, which is kept in contact with the fixed electrode by means of a spring and a bell-crank lever until it is separated from it, on the tripping of the magneto, through the impact of the forked rod F on the other arm of the bell-crank lever. A series of sparks passes between the electrodes as a result of the oscillation of the magneto-inductors.

This method of ignition gives admirable results; and it is particularly applicable where one igniter, only, is to be used. If more than one igniter is necessary, there is required a separate magneto for each igniter. The action of this apparatus is noisy; and if several are in use, the noise becomes quite objectionable. Also, it is not applicable with high speeds of rotation. Its great advantage is that it gives an equally good spark at all speeds, and that it does away with the necessity for supplementary sources of electricity. When there are several igniters, or when the speed is very high, the rotary forms of the magneto are more satisfactory.

Jump-Spark Ignition

Characteristics of High-Tension Method. The make-and-break electrical method of ignition, hitherto described, requires in every case that there shall be a movable electrode subjected to the high temperature of the cylinder. This arrangement, although carried out with success on nearly all stationary gas engines, has

inherent objections. The difficulty of keeping the ignition in working order grows as the size of the engine decreases, as its speed increases, and also with the multiplication of cylinders. In automobile and motor-boat engines, it is particularly desirable that a simpler ignition method should be used. This is accomplished by the *jump spark*.

If an electric circuit is complete except for a small air gap (of, say, .1 inch), and if the electromotive force in the circuit is continuously increased, it will at last reach such a magnitude that it will be able to overcome the resistance of the air gap, and a *spark*, or *electric arc*, will spring across the gap. As the resistance of an air gap is very high, a considerable electromotive force is always necessary, and consequently this method is spoken of as a *high-tension* method. The moment the spark passes between the electrodes, the resistance of the air gap is reduced enormously, so that the high tension is necessary only for starting the spark, not for keeping it up. The resistance increases with the width of the air gap. With a make-and-break contact, the current is passing before the electrodes separate; and the spark which passes from one electrode to the other encounters little resistance until they are separated a considerable distance, because the continuous spark keeps the resistance low. When, however, the spark is finally interrupted by the increase in the air gap, it can no longer jump the gap, even when the electrodes approach very near to each other, because it is only a low-tension current that is used for make-and-break ignition.

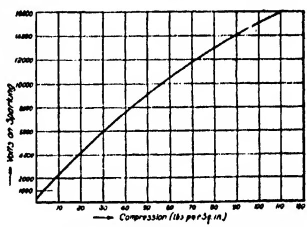


Fig. 131. Curve Showing Voltage Required for Sparking across a .02-Inch Gap with Various Compressions

With a constant air gap, the resistance to the passage of the spark increases with the pressure of the air which surrounds the electrodes. In gas engines the spark is always required to pass after the charge has been compressed; and, consequently, a considerable voltage is necessary. The actual voltage required is shown graphically in Fig. 131 for the small air gap of .02 inch; with 80 pounds compression, over 12,000 volts is necessary. For larger air gaps, the

necessary voltage is still greater. A $\frac{1}{16}$-inch gap at atmospheric pressure requires 10,000 volts. Probably about 100,000 volts is necessary on many spark plugs.

An ordinary cell, primary or storage, will give about two volts, so that it is obviously impracticable to get the desired electrical pressure by putting sufficient cells in series. The magnetos described previously also give low voltages, say, 100 to 200 volts as the maximum pressure. In order to use these sources of electrical power for jump-spark ignition, an induction coil must be used to transform this low-tension current into the desired high-tension current.

Induction Coils Give High Voltage. *General Theory.* When a current flows through a coil of wire, magnetic lines of force are set up surrounding (interlinking) the coil. Conversely, if magnetic lines of force are made to cut a coil, an e.m.f. will be set up in the coil, whose magnitude, as already explained, depends on the rate of change of the interlinkages. If the current flowing through a coil ceases suddenly, the magnetic lines of force cease also—that is, there is a sudden change of the interlinkage; and, as a result, a current will be induced, just as if the magnetic lines had been due to an outside magnet which was suddenly removed. The induced current is in the same direction as the current that was interrupted. This phenomenon is called *self-induction*. The self-induction is greatly increased if there is a bundle of soft-iron wires inside the coil of wire, as this causes a greater concentration of the lines of force and increases the interlinkage. The ordinary spark coil which is used in make-and-break circuits, with battery for source of energy, is built on this principle. When magnetos are used for the generation of the electrical energy, the armature acts as a spark coil, so that no other spark coil is necessary. The effect of the spark coil is to increase the electromotive force at the instant when the current is interrupted; and, when this interruption is due to the actual breaking of the circuit, to cause a spark to jump across the gap formed. This is what takes place in the make-and-break circuit.

Primary and Secondary Coils. If two coils of wire are wound on the same core of iron wire, and if one of these coils, the *primary coil*, is connected to a source of current, and the other, the *secondary coil*, is closed upon itself, then the same number of lines of force will cut both coils, but the interlinkages will depend in each coil on the num-

ber of turns in the coil. If the secondary coil has one hundred times as many turns as the primary coil, the interlinkage with the secondary coil will be one hundred times greater than with the primary coil; and, consequently, on the interruption of the primary current and the disappearance of the magnetic lines, the rate of change of interlinkage will be one hundred times as great as in the primary coil. The pressure of the current induced in this way in the secondary coil, can be made as high as desired by increasing the number of turns of the coil. The action of the one coil on the other is called *mutual induction*.

Use of Condenser. The voltage in the secondary coil depends not only on the number of turns, but also on the rate at which the magnetic lines of force threading the coil are broken down. This latter depends on the rate of disappearance of the current in the primary coil. Now, is it not possible to stop the flow of current in the primary coil instantaneously, with an induction coil made up of the elements mentioned above. It will be found, on trying it, that only feeble sparks will be given by the secondary coil. The trouble arises from the self-induction of the primary coil, which, as described above, tends to keep the current flowing after the circuit has been broken, and causes a spark to jump across the broken primary circuit. The spark in the primary circuit will be found to be even larger than the spark that can be obtained in the secondary circuit; and it not only does no good, but on the contrary is most harmful, as it quickly burns away the contact points in the primary circuit. To remedy this trouble, the self-induction of the primary coil must be overcome; and this is accomplished by means of a *condenser*.

A condenser consists of a large number of thin sheets of tinfoil separated from one another by sheets of paraffined paper or other insulating material. If the sheets of tinfoil are considered as numbered in order, all sheets of even number are connected together and to one terminal of the condenser; and all sheets of odd number are connected to the other terminal. The condenser is then connected across the break in the primary circuit. A condenser constructed in this way has capacity for holding or retaining an electrical charge. When the primary circuit is broken, the self-induced current, instead of forcing its way across the gap, finds its path of least resistance into the plates of the condenser, and goes there and is retained. If the capacity of the condenser is sufficient, the current in the primary

will die down instantly, and consequently a high pressure will be induced in the secondary coil.

Revolving Contact Timer in Primary Circuit. The making and breaking of the primary circuit for jump-spark ignition is brought about by a revolving-contact timer which replaces the tripping device of the low-tension system, and which is the only moving part that is necessary. A timer such as that described earlier and shown in Fig. 128, is a common type for this purpose; and it serves to make and break the primary circuit in four separate induction coils, the secondary coils of which are connected to the spark plugs of four cylinders.

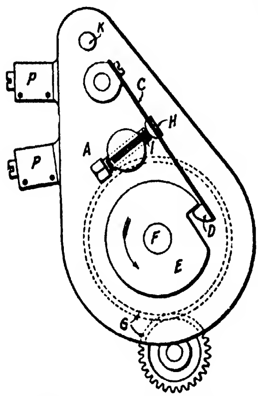


Fig. 132 Diagram of Mechanical Vibrator

Vibrator. With an induction coil as described, this would give one vigorous spark whenever the timer breaks a contact. Such an arrangement is common on bicycle motors. It is desirable, however, to have a number of sparks passing between the electrodes of the spark plug, so as to insure greater certainty of ignition than is possible with a single spark. This can be accomplished by having a rapid succession of makes and breaks of the primary circuit at the time when ignition is desired. The device for effecting this is called a *trembler* or *vibrator* or *buzzer*. The trembler or vibrator may be either *mechanically* or *magnetically* actuated, the latter method being that in most general use at present. The mechanical vibrator is but little used.

Mechanical Vibrator. One of the best known forms of mechanical vibrators is shown in Fig. 132. *H* and *I* are the contact points through which the primary current passes. On the tripping of the spring blade *C* by the cam *E*, the spring is set in rapid vibration, and consequently, there is a rapid succession of makes and breaks at the contact points. The timing is varied in this device by rotating the

base plate A on which the contacts are supported, about the shaft F. The cam E is driven by gears from the engine shaft at one-half the engine speed. PP are the primary terminals, and are connected with H and I, respectively. A clockwise rotation of the plate makes the ignition earlier. The mechanical trembler has been generally discarded because of the breaking of the spring blades, the burning-out of the contact points and other troubles.

Fig. 133. Contact Maker of Atwater Kent Unisparker
Courtesy of Atwater Kent Manufacturing Works, Philadelphia, Pennsylvania

Atwater Kent Contact Maker. A device similar to Fig. 132 is shown in Fig. 133. The hardened-steel rotating shaft in the center has as many notches as there are cylinders, and rotates to the right—clockwise—at half engine speed. Each notch in turn engages the hook-shaped lifter E, drawing it to the right till a certain point is reached. Then the notch releases the lifter, which flies back under the pull of the small coil spring. As the lifter returns, it rides up on the rounded part of the shaft, striking the pivoted hammer, which is located between the lifter and the contact spring, and causing the hammer to force the contact spring D, for an instant, against the contact screw C. The motion is very rapid—the hammer and contact spring appearing to remain stationary. With this device, only one spark coil is needed even for a multi-cylinder engine, the contact being made and broken in the primary by the contact maker and the secondary current sent to the cylinders by a distributor mounted on the same shaft as the contact maker. The mechanism is

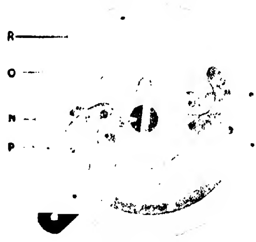


Fig. 134. Automatic Spark-Advance Device of Atwater Kent Unisparker

such that the duration of contact is constant regardless of speed, and is only long enough to build up the current in the spark coil. The moving parts are extremely small and light and, therefore, the inertia effect is reduced to a minimum.

Fig. 134 shows a device which is used in connection with the contact maker just described and is mounted on the same shaft and in the same case. It is a centrifugal governor which advances the spark time as the speed increases. The rotating shaft is divided, and as the governor weights expand they rotate the upper part of the shaft in its own direction of rotation, thus making and breaking contact earlier than at slow speed. By the use of this device the

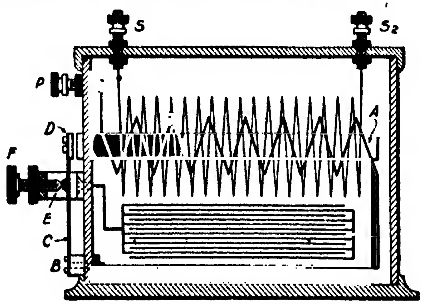


Fig. 135. Induction Coil and Trembler

spark is, automatically, properly timed to correspond with the engine speed.

Action of Induction Coil. It is now the general practice to have a magnetic buzzer or vibrator as part of the induction coil. An ordinary induction coil is shown in Fig. 135. The primary winding leads from the terminal *P*, around the soft-iron core *A*, to the metal plug *B*. The secondary winding leads to the two terminals S_1, S_2. A flat steel spring *C* is fastened to the plug *B*, and has riveted at its free end the soft-iron armature *D*. In the normal position of the spring *C*, the armature *D* is separated a short distance from the armature core *A*, and the platinum-tipped contact point on the back of the spring touches the similar contact point at the end of the

adjusting screw F. F is connected through the battery and timer with the terminal P.

When current is sent through the primary circuit, the core A is magnetized, attracts the armature D, and breaks the contact at E. This interrupts the current in the primary circuit, and with the aid of the condenser induces a powerful current in the secondary. As soon as the current in the primary winding ceases, the core loses its magnetism, and the armature D returns to its normal position, re-establishing the current in the primary. The cycle of operations then recommences and continues so long as current is supplied to the primary coil. The time required for one make-and-break—that is, for one complete vibration of the spring—is generally less than $\frac{1}{100}$ of a second. The rapidity can be varied by adjusting the contact screw F—which is held in place by the locknut shown—the frequency increasing as the screw is advanced.

It is not desirable to have a very light contact between F and the spring, because, in that case, a very small force suffices to break the contact, and, consequently, the primary circuit will be broken before the current has reached its maximum value. This results in a weak magnetic field, and, therefore, in small inductive effect and weak spark in the secondary.

Induction coils are applied to engines which frequently have very high speed of rotation—1000 revolutions per minute, or more. With a trembler making 100 vibrations per second, and an engine making 1000 revolutions per minute, the crank will have turned through an angle of 60° between successive sparks. It is obvious that the interval of time between successive sparks is altogether too great in this case, since, if the first spark does not effect the ignition, the second spark will come far too late to give efficient results. It is desirable, then, for high-speed engines, to make the vibration more rapid. The natural period of vibration of the ordinary hammer vibrator depends on the dimensions of the spring and the mass of the armature. The spring, however, cannot be shortened below certain limits, as that increases its stiffness too much, intensifies the magnetic force required to move it, and, therefore, demands a larger armature.

For best effect—that is, to get a greater induced current—the break in the primary circuit should be made more suddenly than is

accomplished by the ordinary vibrator. With the ordinary vibrator, the circuit is broken as soon as the spring begins to move—that is, while the velocity of the spring is still low. To accomplish a more sudden break of contact, the moving part of the vibrator may be made in two parts, as in Fig. 136. The hammer or armature, which is magnetically attracted to the core, does not carry any contact point, but carries, instead, a button which, after a certain movement of the hammer, strikes the contact spring and breaks the primary current flowing through the contact spring to the contact screw. When the contact is broken, the hammer is in the middle of its stroke, and is moving with considerable velocity. The result is a rapid break. The substitution of the thin hammer for the heavier iron armature Fig. 135, permits higher speed, as the inertia effects are less.

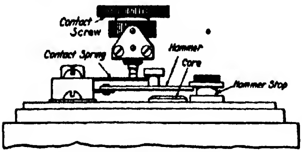


Fig. 136. Vibrator of Splitdorf Coil
Courtesy of Splitdorf Electrical Company, Newark, New Jersey

The vibrations per second of the trembler vary in the principal coils from about 100 to 400. They are generally designed for from 4 to 6 dry-cell batteries, or a 3- or 4-cell storage battery. A good coil requires about .2 to .25 ampere when in use on a single-cylinder engine. With low compression in the engine, and the resulting comparatively low voltage required in the secondary coil, the pressure of the contact on the trembler spring can be made very slight, as it is not necessary to develop the full current in the primary coil. With high-compression engines, the pressure of the contacts must be increased, and the use of current will increase correspondingly.

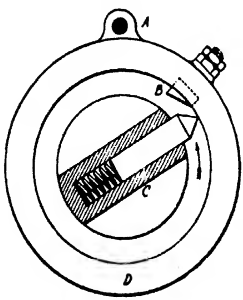


Fig. 137. Snap-Off Timer

There is provided in all coils a safety spark gap to prevent over-

straining of the insulation in case a current of abnormally high voltage is sent through the coil. The current will pass through this gap if the spark plug is taken off, in which case there is no small air gap in the circuit.

Timers with Separate Induction Coil. If there are several igniters on an engine, they may be served either by a separate induction coil for each igniter, or by a common induction coil for all igniters. With a separate induction coil for each igniter, and one source of electrical energy, a timer or *primary commutator* must be used, rotating in synchronism with the engine and sending the primary current to the different coils in succession at the desired times. One form of such timer has been shown already in Fig. 128. Other forms are shown in Figs. 137 and 138. With the *snap-off timer*, Fig. 137, the pressure of the spring insures a good contact between the rotating contact piece C and the fixed contact B; and the ending of the contact is so abrupt that it may cause a spark in the secondary coil, even if the vibrator refuses to act. Only one contact B is shown, fastened to the non-conducting case D; but there will be as many contacts around the periphery of the timer as there are cylinders.

Fig. 138. Connecticut Roller-Contact Timer for Four Cylinders
Courtesy of Connecticut Telephone and Electric Company, Meriden, Connecticut

In those cases where the noise and wear of this type of timer are objectionable, the *roller-contact timer*, Fig. 138, may be used. In both cases, by the simple device of rotating the external casing through the desired angle, the times of all the contacts can be advanced or retarded simultaneously and by the same amount.

Distributors. With separate induction coils for the separate cylinders, the timing of ignition will not be quite the same in each cylinder, although the timer contacts occur at exactly the proper intervals.

This results from the fact that it is not practicable to adjust the vibrators of the coils so that all have the same period of vibration. Consequently, the ignition lag will be different in the different cylin-

ders; this is why it is important to endeavor to adjust all the vibrators till they give the same note. By the use of one coil for all cylinders, this trouble can be remedied, and we get the so-called *synchronous* system.

If it is desired to use but one induction coil for several cylinders, a timer is still necessary to send current to the primary coil at those times when ignition is desired; but a distributor, or secondary commutator, is also necessary, to send the high-tension current generated in the secondary coil to the proper spark plug. The very high voltage of the secondary circuit renders the construction of a distributor much more difficult than the construction of a timer. In principle and in method of action, they may be precisely similar; but it is necessary to give extraordinary care to the insulation of the distributor, while with the timer this gives but little trouble. The distributor is generally mounted on the same shaft as the timer, or is geared directly to it.

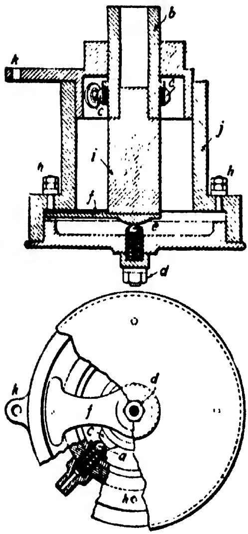


Fig. 139. Combined Timer and Distributor

On account of the high tension in the secondary circuit, it is not necessary that the revolving arm of the distributor should actually touch the insulated fixed contacts; if current is being generated while the revolving arm is close to one of the contacts (say, $\frac{1}{16}$ inch away), a spark will jump across the gap. By the use of a glass top to the distributor, the action of the coil can be observed. A combined timer and distributor is shown in Fig. 139, the timer being above, the distributor below. The primary current enters through the steel ball *a*, which comes in contact with cams *c* on the rotating sleeve *b*. The secondary current enters at *d*, and goes

through the steel ball *e* to the brass strip *f*, and thence to the base of one of the binding-posts *hh*. Insulation is effected by having the post *i* and the casing *j* of hard rubber. Advance or retardation of the spark is effected by the rotation of the case *j* through the arm *k*.

High-Tension Magnetos. If a magneto is used to supply current for jump-spark ignition, it is called a high-tension magneto. It may be precisely the same as the low-tension magneto described previously, generating a low-tension current which goes to a separate induction coil; or it may have the secondary coil wound on the armature of the magneto, so that the magneto acts not only as a current generator but as induction coil also. The latter is the common method. Since one magneto is all that is necessary for several

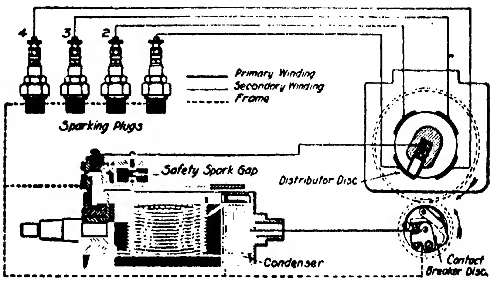
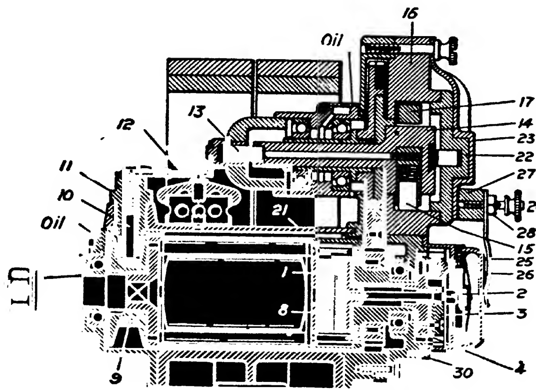


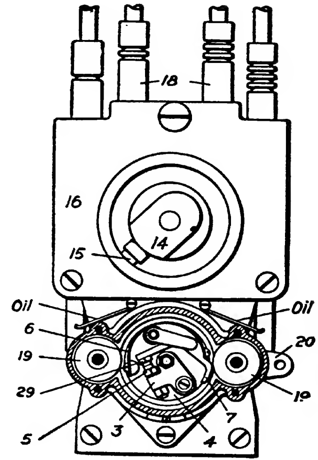
Fig. 140. Wiring Diagram for Bosch High-Tension Magneto

cylinders, it is usual to make the distributor an integral part of a high-tension magneto. A timer, interrupter, or circuit breaker is necessary to break the primary circuit rapidly at the desired time, so as to give a good induction effect.

High-Tension System for Four-Cylinder Engine. The general arrangement of a high-tension magneto ignition system for a four-cylinder engine is shown in Fig. 140. The primary and secondary windings of the magneto are continuous with each other. One end of the primary winding goes to the armature core; the other end goes to a contact breaker which, normally, short-circuits the primary coil, but which, at the moment of sparking—when the movement of the armature is such as to give a vigorous



Longitudinal Section



Rear View

Fig. 141. Bosch High-Tension Magneto

1—Brass Plate; *2*—Contact-Breaker Screw; *3*—Platinum Screw Block; *4*—Contact-Breaker Disk; *5*—Long Platinum Screw; *6*—Contact-Breaker Spring; *7*—Contact-Breaker Lever; *8*—Condenser; *9*—Slip Ring; *10*—Carbon Brush; *11*—Carbon Holder; *12*—Connecting Bridge; *13*—Contact Carbon; *14*—Rotating Distributor Piece; *15*—Distributor Carbon; *16*—Distributor Disk; *17*—Metallic Segments; *18*—Contact Plug; *19*—Fiber Roller; *20*—Timing Lever; *21*—Dust Cover; *22*—Cover; *23*—Triangular Clamp; *24*—Nut for Switch Wire; *25*—Spring for Fastening Brass Cap; *26*—Brass Cap; *27*—Brass Block for Fastening Spring of Brass Cap; *28*—Fixing Bolt; *29*—Short Platinum Screw; *30*—Stop Screw for Timing Lever

current—breaks the circuit suddenly, and consequently induces the necessary current in the secondary. The armature is of the usual rotating type, running at the same speed as the engine and giving two electrical impulses per revolution. Hence the contact breaker is arranged to break contact twice per revolution, giving two electrical impulses in the secondary circuit per revolution of the engine. A condenser is connected across the circuit-breaker gap in the primary circuit. The secondary winding is grounded at one end by being made continuous with the primary, and at its other end goes to an insulated ring at the left, and then, through a brush, to the distributor. The safety spark gap minimizes the probability of injury to the insulation of the secondary coil from excessive voltages. The distributor arm is geared to the contact breaker, and revolves at one-half its speed; that is, it makes one revolution for two revolutions of the engine. The rotating arm makes successive contacts with each of the four insulated segments during a revolution, and consequently sends current to the spark plugs.

Variation in the time of ignition is effected by varying the time of interruption of the primary circuit.

Construction of Magneto. The constructive details of this magneto are shown in Fig. 141. The end of the primary winding is connected to the brass plate *1*. In the center of this plate is screwed the fastening screw *2*, which serves, in the first place, for holding the contact breaker in its position, and, in the second, for conducting the primary current to the platinum screw-block *3* of the contact breaker. Screw *2* and screw block *3* are insulated from the contact breaker disk *4*, which has metallic connection with the armature core. The platinum screw *5* goes through the screw block *3*. Pressed against this platinum screw by means of a spring *6*, is the contact breaker lever *7*, which is connected to the armature core and therefore with the beginning of the primary winding. The primary winding is therefore short-circuited as long as lever *7* is in contact with platinum screw *5*. The circuit is interrupted when the lever is rocked. A condenser *8* is connected in parallel with the gap thus formed.

The end of the secondary winding leads to the slip ring *9*, on which slides a carbon brush *10*, which is insulated from the magneto frame by means of the carbon holder *11*. From the brush *10*, the secondary current is conducted to the connecting bridge *12*, fitted

with a contact-carbon brush *13*, and through the rotating distributor piece *14*, which carries a distributor carbon *15*, to the distributor disk *16*.

In the distributor disk *16*, are embedded metal segments *17*. During the rotation of the distributor carbon *15*, the latter makes contact with the respective segments, and always connects the secondary current with one of the contacts.

The contact breaker is fitted into the rear end of the armature spindle, which is bored out and provided with a keyway. The contact breaker is held in position by screw *2*. The short-circuiting and interrupting of the primary circuit is effected by means of the contact-breaker lever *7*, on the one hand, and the fiber rollers *19*, on the other. As long as the lever *7* is pressed against the contact screw

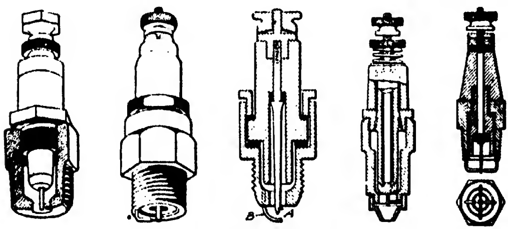


Fig. 142. Typical Forms of Spark Plugs

5, the primary circuit is short-circuited, and the rocking of the levers by the fiber rollers *19* effects the break of the primary circuit; at the same moment ignition takes place. The distance between the platinum points, when the lever is lifted on the fiber rollers, must not exceed .5 millimeter (approximately $\frac{1}{50}$ inch). This distance may be adjusted by means of the screw *5*.

Spark Plugs. The part of a jump-spark system that is most likely to give trouble is the spark plug itself. The spark plug contains two electrodes, with an air gap which is usually between $\frac{1}{32}$ and $\frac{1}{16}$ inch. One of these electrodes is grounded, the other is insulated as perfectly as possible. The difficulty is in keeping the insulation good under the very high voltages of the secondary circuit. Not only must the insulation be electrically good, but it must also be gas-tight under the high pressures existing in gas-engine cylinders.

Some common forms of spark plugs are shown in Fig. 142. They all consist of three fundamental parts—the *plug body*, which screws into the engine cylinder and is thereby grounded; the *insulated electrode*, and the *insulating body*. The insulation is effected by the use of either porcelain or mica. The former is the more brittle, and, as it is subjected to a high temperature inside the cylinder and a low temperature outside, the unequal expansion resulting is liable to crack it; moreover, it is not well adapted to withstand rough usage.

Mica insulation is built up of washers of sheet mica, generally without any cement between the washers. It is free from the general objections to porcelain, and is being largely used. With either form of insulation, however, trouble is likely to arise from the sooting of the plug—that is, from the deposit of carbon on the plug. This deposit is most likely to form on the surface of the insulator, and forms a conducting bridge from the insulated electrode to the plug body. Even if the spark plug works satisfactorily when tried in the open air, it may not work in the cylinder, as the greater resistance which the compressed gases offer to the jumping of the spark may cause the current to go over the surface of the insulating material if this is not clean. To increase the resistance to such leakage of the current, the surface of the insulator is often made greater by corrugations.

In Fig. 142, the first plug is a *closed-end plug*. Some of the charge is compressed into the plug; and being the part of the charge that is first ignited, it expands and rushes out of the enclosed space so violently as to prevent carbon deposit. The second and third are of the *open type*. The fourth plug is another example of the closed type; its insulation consists of a mica tube inside a porcelain tube. The porcelain is held in place, gas-tight, by an accurate taper-ground joint without packing. The spring on top takes up heat expansion of the porcelain. The fifth plug is of the open type, with four grounded electrodes around the central electrode; there are two porcelain bushings around the insulated electrode.

The electrodes are sometimes of platinum, but more commonly of nickel steel, which resists oxidation nearly as well as platinum.

Comparison of Ignition Systems. A comparison of the magnitude and duration of the current flowing in the various methods of

ignition is given by the curves in Fig. 143. With make-and-break ignition, Fig. 143*a*, the current increases from the time the contact points are brought together till the circuit is opened; then the arc is drawn, and lasts while the coil, which is still receiving current from the source of electricity, discharges its magnetic energy; this time may be about five-thousandths of a second. With high-tension ignition without a vibrator, Fig. 143*b*, the only difference of the primary current curve from Fig. 143*a* is that by the action of the condenser the primary break is instantaneous. The resulting secondary-induced current is of smaller amount, rises instantaneously to its maximum value, and lasts about one-thousandth of a second. With a vibrating coil, Fig. 143*c*, having the same duration of closing of the primary circuit by the timer as in the previous case, there is seen to be less energy for each spark in the secondary, as the current does not build up as high in the primary during the shorter contacts.

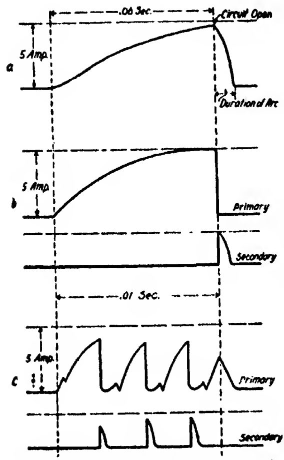


Fig. 143. Curve Showing Comparative Magnitude and Duration of Current Flowing with Various Systems of Ignition

The make-and-break system of ignition gives a hotter spark and one of longer duration than is obtainable with jump-spark ignition, and hence gives more effective ignition; it is used almost exclusively in large engines. This system is, electrically, most simple, but mechanically it is complicated. The jump spark, on the other hand, is mechanically simple, while the electrical system is complex. The mechanical simplicity of the jump-spark system has led to its practically exclusive use in automobile and motor-boat engines; moreover, it is better adapted to high speeds of rotation.

ENGINE DETAILS

GOVERNING

Functions of a Governor. The governing of an engine means the control of the power which it is developing, so that its speed is maintained practically constant. If the engine develops more power than is required, the engine will speed up; if the power delivered to the crankshaft is less than the resistance there, the engine will slow down. The governing of a gas engine, like that of the steam engine, is effected by utilizing small variations of engine speed resulting from change of engine load. The controlling mechanism, or the governor proper, does not differ from that used on the steam engine. If the work that must be done by the governor proper, or regulator head, in moving the governor valve mechanism to correspond with changes in the load is light—as is the case in small and medium-power engines—a fly-ball or an inertia governor is generally used, as seen in Figs. 144 and 153, respectively. If the valve mechanism is heavy and apt to stick because of impurities in the gas—as in a large engine—a regulator head, such as is shown in Fig. 145, is employed. The way in which the governor mechanism controls the work done by a gas engine is very different from that employed in a steam engine. There are two general methods in use in gas engines for varying the power—one by varying the number of explosions or impulses per minute, which is

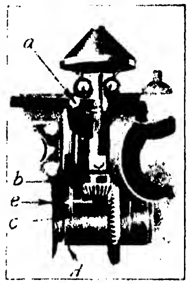


Fig. 144. Otto Engine Governor
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

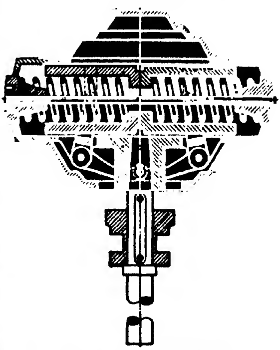


Fig. 145 Hartung Regulator Head
From Güldners' "Verbrennungskraftmaschinen"

known as the *hit-and-miss* system; and the other by varying the magnitude of the impulse while keeping the number per minute constant, which may be called the *variable-impulse* system.

Hit-and-Miss System. *Method.* The omission of the explosion or impulse can be obtained in several ways. The most common method is to keep the gas-admission valve closed so that air alone is taken in during the admission stroke, and, consequently, there is no explosion. A method of accomplishing this is to be seen in Fig. 144, in which a loaded centrifugal governor is shown driven by bevel gearing from the camshaft. In the position shown, the gas-admission cam *d* will come under the reel *c*, and will start to lift it at the beginning of the admission stroke. The reel *c* is loose on a spindle at the end of the horizontal lever *e*, and the vertical rise of the spindle, due to the action of the cam, opens the gas valve by a system of levers not shown in the figure. If the engine speeds up, the rise of the governor balls raises the sleeve on the governor spindle, lifts the horizontal arm of the bell-crank lever fulcrumed at *a*, and shifts the forked end *b* of the vertical arm to the right, carrying the reel *c* with it, so that the cam no longer engages it and no gas is admitted. When the speed comes down to normal, the reel is moved back, and the admission of gas again takes place.

Disadvantages. The hit-and-miss method is open to the objection that it makes the speed of the engine very irregular at any other than full load. Even at full load, with the Otto cycle and a single-acting cylinder, there is only one motive stroke or impulse in four strokes, instead of one every stroke as in a double-acting steam engine. If the engine governs by the hit-and-miss method and is running at half-load, half the explosions will be omitted, and there will be but one motive stroke in eight; at one-third load, there is but one motive stroke in twelve; and at quarter-load, one in sixteen. Running at quarter-load, the engine will be speeded up during the motive stroke, and will slow down during the succeeding fifteen strokes, till it gets to normal speed again. The actual variation in speed at low loads can be reduced by use of a heavy flywheel; but with this method of governing, it is too great for use when close regulation is necessary, as, for example, in electric lighting. An incidental advantage of this method is that, during idle cycles, the cylinder is flushed out by the *scavenging* charge of air, making the next explosion more powerful.

For loads approaching full load, where the number of misses is small, the explosion directly following a miss will be more powerful than the average, but those succeeding it, before another miss, will each be weaker than the first one, due to the fact that when there are no misses the gases remaining in the clearance are not scavenged out, and dilute the charge with inert gases. At very low loads the number of misses greatly exceeds the number of explosions and, for that reason, the first explosion following a miss will be weaker than the average, due to the fact that the cylinder has been cooled off by the large number of scavenging charges of cool air, and, therefore, the first charges are somewhat slow-burning. These facts add to the irregularity of governing and although, theoretically, this system gives the best fuel consumption, practically, the efficiency is extremely variable.

In engines which have an automatic admission valve, the omission of an explosion is sometimes effected by the action of the governor in keeping the exhaust valve open throughout the cycle. The free communication between the cylinder and the outside, through the exhaust valve, prevents the formation, during the admission stroke, of the vacuum necessary to open the admission valve. Consequently, so long as the exhaust is open, the admission valve will remain closed; the cylinder will contain only products of combustion; and no explosion can occur. This system has the drawback that the cylinder is not scavenged at all.

Variable-Impulse System. The amount of work done in a given gas engine depends on the strength of the charge (*qualitative governing*), on its amount (*quantitative governing*), on the timing of the ignition, and on several other factors. The engine can be governed by the variation of any one of these systems or a combination of any two; and the three specifically mentioned are all in regular use for this purpose.

Qualitative Governing. If the governing is effected by varying the strength of the charge, the control has to be such that the mixture is always an explosive one. With each kind of gas used in an engine, there are both higher and lower limits to the amount of air with which it may be mixed if it is to remain an explosive mixture. If the ratio of air to gas should be outside these limits, the mixture sent to the exhaust would be unburned, and valuable gas would be lost. It

naturally follows that if the engine goes above normal speed when admitting the weakest explosive mixture, the power of the engine has to be further reduced by omitting the admission of gas entirely. In Fig. 146 is shown a device for governing in the manner just described. The governor d is driven from the camshaft c through the bevel gears shown. The gas is admitted by raising the end of the lever, on which is a reel b similar to c in Fig. 144. The sleeve a is free to slide on a feather on the camshaft c, its exact position being controlled by the governor through the bell-crank lever e. On the sleeve a is a series of cams of the same throw, but of different circumferential lengths. The duration of the admission of gas is varied by shifting the sleeve so as to bring different cams into engagement with b. In the position shown, the engine is above normal speed, the sleeve is at extreme position to the right, and

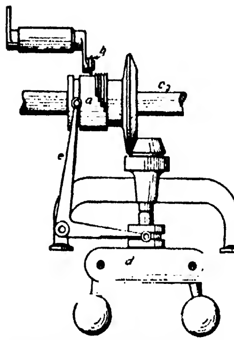


Fig. 146. Sketch Showing Typical Governor Operating Under Qualitative System

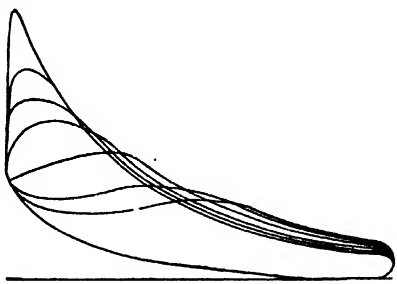


Fig. 147. Indicator Cards Taken at Different Loads; Qualitative Regulation
Courtesy of Buckeye Engine Company, Salem, Ohio

no gas is being admitted. As the speed of the engine falls, the sleeve travels to the left, admitting gas for a definite period for each engine

speed. With full load on the engine, the reel engages with the longest cam, and the strongest mixture is admitted to the cylinder.

With this method of governing, the same amount of the mixture is always taken into the cylinder, and, consequently, the pressure at the end of compression is always the same. The explosion, however, becomes weaker as the mixture is "leaner", and requires a longer time for its completion. A comparison of the areas of Figs. 8 and 10, pages 22 and 23, shows the effect of a weaker mixture on the power of the engine. The manner of applying this method of regulation to a

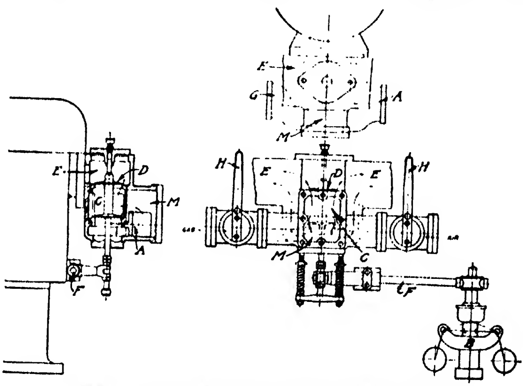


Fig. 118. Diagram Showing Mixing Valve of Westinghouse Throttle Governor
Courtesy of Westinghouse Machine Company, Pittsburgh, Pennsylvania

large gas engine is described on page 124, and the details of the construction of the governor mechanism are shown in Figs. 65 and 66, pages 122 and 123. The effect of a considerable fluctuation of load on the indicator cards, with this system of regulation, is shown in Fig. 147.

Quantitative Governing. It is found, in practice, that there is a certain strength of the explosive mixture which gives the most economical running of the engine. It is obviously desirable to run the engine with a mixture of this strength; and that can be done when a hit-and-miss governor is used. When it is desired to have an impulse every cycle, a constant strength of mixture can be maintained if the power of the engine is controlled by varying the amount of the

mixture taken in. Two methods of obtaining this system of regulation are in practical use: the throttling method, in which the mixture proportions are kept constant and the charge is throttled down throughout the suction stroke by governor-operated valves; and by the cut-off method, in which the incoming charge is completely cut off by the governor at some point in the suction stroke, the charge expanding for the rest of the stroke.

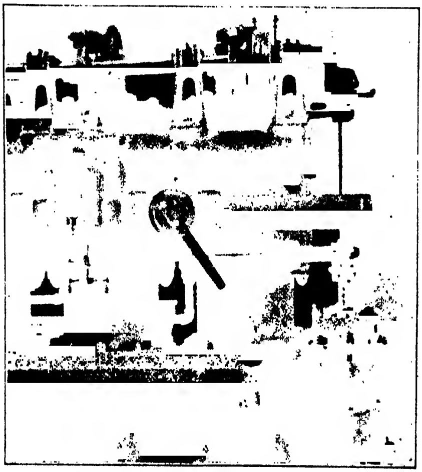


Fig. 149. Governor of Westinghouse Gas Engine
Courtesy of Westinghouse Machine Company, Pittsburgh, Pennsylvania.

Throttle Governing. An example is shown in Figs. 148 and 149 of the actual mechanism used for this purpose. Gas from the passage *G* enters the mixing chamber *M* where it meets air entering from *A* through a similar passage. The mixture flows from the mixing chamber to the governor-valve chamber *C*. The governor valve *D* is a double-beat poppet valve, so that the mixture flows from the governor-valve chamber at the middle of the valve to the engine inlet passage *E* through both the upper and lower valves, as shown

by the arrows. The relative amounts of gas and air are regulated by the two levers H H, which operate the two plug valves in the gas and air passages. With the two levers in constant positions, the areas for admission of gas and air to the mixing chamber M will be fixed, and consequently the strength of the mixture will be constant. The actual amount of the mixture entering the cylinder is controlled by the governor B, through the governor lay shaft F, so that the governor valve D is almost closed—the mixture is throttled—when the speed increases.

This method of governing permits a perfect adjustment of the work done in the cylinder each cycle, and consequently gives more uniform speed of the engine than any of the methods so far described. Another method of accomplishing it is shown in Fig. 150. The air and gas pass through a mixing valve which controls the proportions according to the power demand, before it reaches the throttle valve.

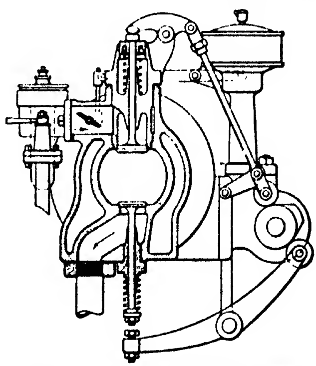


Fig. 150. Throttle Governor of Crossley Engine
Courtesy of Crossley Gas Engine Company, Manchester, England

The throttling of the mixture imposes extra work upon the engine during the admission stroke, as the piston has to move out with a vacuum behind it. At the end of the admission, the pressure in the cylinder will be less and less as the load on the engine becomes smaller, and, consequently, the pressure in the cylinder at the end of compression is less as the load decreases. With decreased compression, the combustion of the mixture is slower This is well shown in Fig. 151, which gives a series of indicator cards taken at different loads from an engine using a strong mixture and a throttling governor.

The throttling governor valve shown in Figs. 148 and 149 has the inherent disadvantage that the mixture is made at the valve and, consequently, the passage from the throttle valve to the inlet valve

is full of explosive mixture at all times, so that there is a considerable volume of gas to explode in the inlet passages if a back fire should occur. The valve shown in Fig. 150 has the additional disadvantage that the pull on the valve, due to the engine suction, is unbalanced and tends to pull the governor to a position that does not correspond to the speed of the engine.

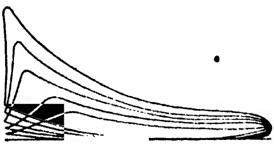


Fig. 151. Indicator Cards Taken at Different Loads; Quantitative Regulation

The valve shown in Fig. 152 overcomes both of these disadvantages, as the pull due to the suction of the engine is balanced by acting in opposite directions on two disks and the gas and air are prevented from mixing by the collar on the valve stem between the gas and air disks. This collar forms a sliding fit in the partition between the gas and air passages. The gas and air are conducted to the inlet valve in separate passages and do not mix until the inlet valve opens (see Figs. 56 and 57 and page 113) and thus, in case of a back fire, the explosion is confined to the small volume of mixture contained in the inlet-valve cage.

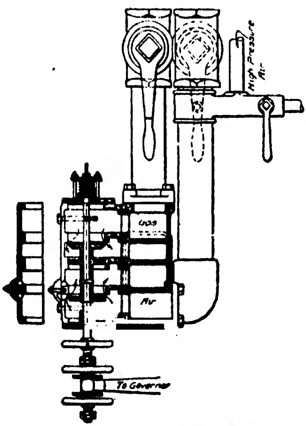


Fig. 152. Governor Valve of Bruce-MacBeth Producer-Gas Engine
Courtesy of Bruce-MacBeth Engine Company, Cleveland, Ohio

In the engines shown in Figs. 49 to 68, the throttling method of regulation is almost universally employed.

Cut-Off Governing. Another method of accomplishing quantitati veregulation, as has been already pointed out, is to admit a

mixture at atmospheric pressure for part of the admission stroke only, the duration of the admission being determined by the governor. This method of governing gives an indicator card similar to Fig. 7,

Fig. 153. Governor Bar, Valve, and Cage of Alberger Automatic Cut Off Engine
Courtesy of Alberger Gas Engine Company, Buffalo, New York

page 20. The difference between an engine governing in this way and one governing by the throttling method, is similar to that between a Corliss steam engine and a throttling steam engine. The advantage

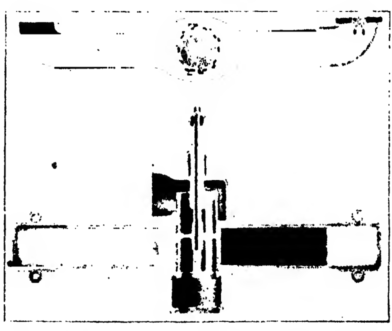


Fig. 154. Sectional Views of Alberger Automatic Cut-Off Valve
Courtesy of Alberger Gas Engine Company, Buffalo, New York

of cut-off governing is in the decreased work done by the engine in drawing the charge into the cylinder. The use of a partial charge, whether obtained by throttling or by cutting-off, permits the expan-

sion of the exploded mixture to a pressure lower than is possible in an engine admitting a full charge and having the same pressure at the end of compression. This is the practical method of obtaining the increased expansion, the advantage of which has been already pointed out in the section on Thermodynamics.

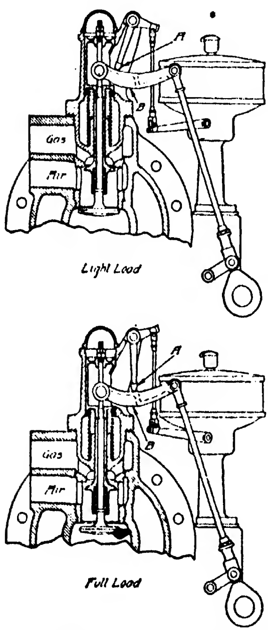


Fig. 155 Sections Showing Variable Admission Governing of Crossley Engine for Light and Full Loads
Courtesy of Crossley Gas Engine Company, Manchester, England

In Fig. 153, the valve-actuating mechanism, and Rites inertia governor of a cut-off system are shown. In Fig. 154 the cut-off valve is shown in position in the inlet manifold. The valve and valve cage each have the same number of ports of equal width, the ports in the valve being longer than those in the cage. The cage is located entirely in the manifold proper, while the valve projects through the manifold into the gas and air inlets on either side. Across the center of the valve, a partition is cast so that the gas enters the manifold above and the air below this partition. By raising or lowering the valve, by means of a thumbscrew, the partition in the valve is raised or lowered and the relative gas- and air-port openings can be varied, while the total opening remains constant, in order to adjust by hand for widely varying gas quality.

The valve is opened and closed by a slight rotary movement imparted to it by the actuating mechanism, which is driven by the equivalent of a variable length crank mounted on the governor bar.

A well-known English method of "variable admission" or quantitative governing is shown in Fig. 155. The governing is effected by varying the lift of the inlet valve by varying the position of the fulcrum *A* on which the radius-lever *B* rotates. The control of the position of the fulcrum by the governor is evident from the drawings. The mixture of air and gas takes place at the admission valve.

When economy is not of the greatest importance, the power of the engine may be controlled by varying the point of ignition. It has been shown already, Fig. 109, page 175, that the power of the engine decreases as the lead of the ignition becomes less. If the ignition occurs after the beginning of the stroke, the lead is said to be negative, and

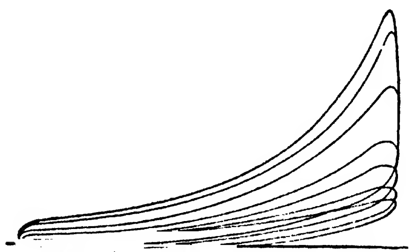


Fig. 156. Indicator Cards Taken at Different Loads—Combined Qualitative and Quantitative Regulation and Spark Automatically Advanced with Decrease of Load
Courtesy of Buckeye Engine Company, Salem, Ohio

the power is greatly decreased. If the lead is increased, Fig. 110, there still results a decrease of power. The control of the power by varying the ignition alone is always uneconomical, but the method is one of extreme simplicity.

With qualitative regulation the spark should be advanced, with decrease of load, over the setting at maximum load in order to allow more time for the combustion, which is slower with the weaker mixture, Fig. 147.

Combined Systems. The best of the modern methods of governing is a combination of the qualitative and quantitative methods. As the power of the engine decreases, the strength of the mixture is decreased till the most economical mixture is reached. For lower

TABLE XIV

Data on Engines Giving Efficiency Curves of Fig. 1..

Curve No.	Type of Engine	Rated b.h p.	Rated r.p.m.	Fuel	Governing System
1	Deutz, single cylinder..	50	200	Illuminating gas	Throttling gas
2	Westinghouse, 3-cylinder vertical...........	100	270	Natural gas...	qualitative Throttling mixture quantitative
3	Deutz....	450		Producer gas..	
4	Güldner, single cylinder	35	220	Producer gas..	Throttling
5	Nürnberg.....	1200	106	Blast-furnace gas......	Throttling gas
6	Swiderski, single cylinder	15	235	Alcohol.	Throttling
7	Deutz, single cylinder.	12	285	Alcohol......	Throttling
8	Diesel.	70	158	Russian kerosene	Cut-off
9	Diesel...	8	275	Kerosene.......	Cut-off
10	Hornsby-Akroyd......	25	202	Kerosene.......	Regulating oil
11	Banki.........	25	210	Gasoline with water injection.	Hit-and-miss

loads, this most economical mixture is kept, but the amount of it admitted to the cylinder is decreased.

The cards shown in Fig. 156 are taken from an engine which has a combination of the qualitative and quantitative methods of regulation, and besides has a governor-actuated ignition advance, so that, as the load falls off, the ignition is advanced to give as nearly constant-volume combustion as the mixture will allow. The operation of the mixing apparatus is as follows: With a maximum load on the engine, the quality of the mixture is such as to give the highest mean effective pressure and the maximum compression possible. When the load falls off, the strength of the mixture decreases until the load on the engine has decreased to about 25 per cent of the maximum load. After this, from 25 per cent of the maximum load down to friction load, the charge is no longer weakened but is varied in quantity by throttling. It will be noticed by examining these diagrams that the compression is practically constant for all loads except the very lowest. It will also be observed that the combustion is much more rapid throughout the range of loads than is the case in Figs. 147 and 151.

A few typical efficiency curves of engines fitted with the

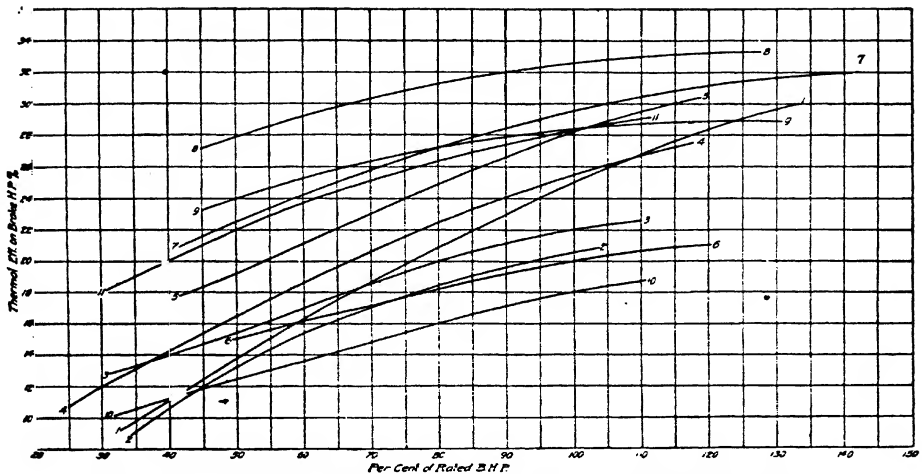


Fig. 157. Efficiency Curves of Engines Fitted with Various Systems of Regulation, for Various Percentages of the Rated Load
From Carpenter and Diederichs' "Internal Combustion Engines"

various systems of regulation, for various percentages of the rated load, are shown in Fig. 157. Table XIV gives the explanatory data corresponding to these curves.

VALVES AND VALVE GEAR

Valves. The inlet and exhaust valves in gas engines are nearly always *poppet* or *mushroom* valves with conical seats similar to those shown in Figs. 44 to 107. The lift is usually about one-fourth the diameter. The exhaust valves are nearly always mechanically operated; the main inlet valves are sometimes automatic. The automatic valve is similar in action to a pump suction valve, and is kept on its seat by a weak spring, opening only when the pressure in the cylinder is sufficiently below the atmospheric pressure to permit the latter to overcome the resistance of the spring. Consequently, the suction or admission pressure in the gas engine is always low when automatic inlet valves are used. The effect is to decrease the amount of the charge taken in, the work done by the engine, and its efficiency; the only advantage is the greater simplicity. Most *small* gas engines have automatic inlet valves.

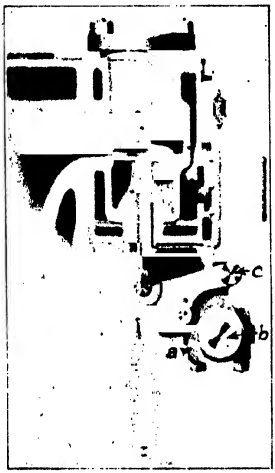


Fig. 158. Valve Motion for Alberger Engine, Valves, 3 to 6 Inches in Diameter
Courtesy of Alberger Gas Engine Company, Buffalo, New York

A positively actuated admission valve is shown in Fig. 158. The valve is lifted by a cam *a* on the side shaft *b*, through the lever fulcrumed at *c*. The valve closes by its own weight, assisted by a spring, and is guided in its motion by a long sleeve. The valve chest is completely water-jacketed. The exhaust valve is located directly behind the admission valve and is actuated by a similar mechanism.

This arrangement of valves and method of actuating them, as applied to larger engines, is shown in Fig. 159. In this arrangement the inlet valve is located directly over the exhaust valve and is actuated through an additional lever and rod as shown. Other positively actuated inlet valves are shown in preceding figures.

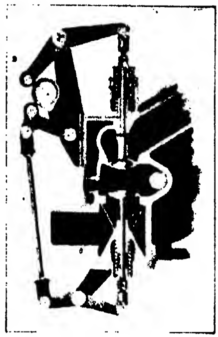


Fig. 159. Valve Motion for Alberger Engine Valves Exceeding 6½ Inches in Diameter
Courtesy of Alberger Gas Engine Company, Buffalo, New York

The pressure in the cylinder when the exhaust valve opens is generally from 25 to 45 pounds above the atmospheric pressure, and the exhaust valve has to be lifted against this pressure. With a mushroom valve 4 inches in diameter, and with 40 pounds pressure per square inch at the end of expansion, there would be a total pressure of about 500 pounds on the valve at the time when it is to be lifted. It is desirable to more favorably take care of the strain on the valve mechanism, and in large engines this is sometimes done by making the exhaust-valve actuating rod lift the valve, rather than push it up; thus putting the rod in tension instead of compression.

Valve Gearing. *Cams and Eccentrics.* The valves are most commonly operated by cams. Cams are preferable to eccentrics for this purpose, because they can be designed to give very prompt opening and closing. In large engines, however, cams soon become noisy, due to wear caused by the heavy total pressures the cams must lift. For that reason, eccentrics are substituted in large engines in place of cams, the slower opening and closing being more than offset by the quietness of running. The cams or eccentrics are mounted upon a *lay shaft*, or *side shaft*, or *camshaft*. The camshaft is driven in different engines either by *spur gears*, *bevel gears*, or *spiral or skew gears*. The spur gear (see Fig. 27) can be used only for parallel

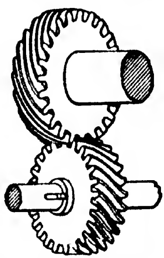


Fig. 160. Type of Spiral Gear

shafts; the bevel gear, for shafts which are in the same plane but are inclined to one another; and the spiral or skew gear, Fig. 160, for shafts which are not parallel and do not lie in the same plane. To reduce the speed of the camshaft, the spur and bevel gears must have the gear on the camshaft twice the size of that on the main shaft. With the spiral gear, there is no necessary relation between the diameters of the two gears and, generally, the gear on the camshaft is smaller than that on the main shaft. The spiral gear has great advantage over the other two in its quietness of operation.

Double-Gear Drive. In large gas engines the lay shaft is sometimes driven by two sets of spiral gears, in which case the only duty of the *first* lay shaft is to drive the *second* lay shaft and the regulator head. The duty of the *second* lay shaft is to operate the valves, igniters, etc., and to that end the eccentrics and cams are all mounted on the second lay shaft. It is a very common practice in such drives to have an odd number of teeth on the spiral drive gears—or, as it is called, a "hunting tooth". By this means the same two teeth, on driver and driven gear, come into contact infrequently and the wear on the teeth is evenly distributed around the gear and the life of the gear thus increased.

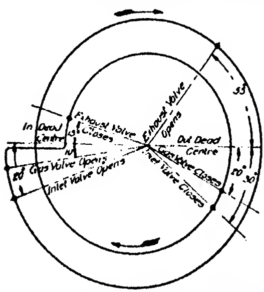


Fig. 161. Valve-Setting Diagram for One End of Large Four-Cycle Double-Acting Horizontal Tandem Producer-Gas Engine

Valve Setting. *Moderate-Speed Engine Practice.* The timing of the various events in an Otto-cycle gas engine depends greatly on the speed of rotation of the engine; the higher the speed, the earlier should be the exhaust and the ignition. For engines of moderate speed, the exhaust valve opens from 30° to 60° before the crank reaches the *out* dead center; and closes when the crank is on the *in* dead center, or shortly after, sometimes as much as 15° after. The admission normally begins 5° to 10° after the *in* dead center, and ceases about 10° to 20° after the *out* dead center, with mechanically actuated valves.

Setting the Valves to Help Scavenging. In some engines, especially in the large ones, the fact that the moving column of exhaust gas has considerable inertia in flowing through the exhaust port is utilized to scavenge the cylinder. Because of this inertia the exhaust gases will keep on flowing out of the exhaust port even if the inlet valve is opened as much as 20° before the *in* dead center. The inertia of the exhaust will also create enough suction on the air to suck some of it through the inlet port, driving the exhaust gases out of the clearance and at least partially scavenging the cylinder. The gas valve is then opened about 10° before the *in* dead center, since there is no danger of the exhaust gases firing the incoming mixture as there is a stratum of air between. The inertia acquired by the mixture during the suction stroke is similarly utilized to obtain more complete cylinder filling by keeping the inlet valve open until as late as 30° after the *out* dead center. The advance of the ignition depends largely on the kind of ignition employed; it averages about 30° with electrical ignition.

In Fig. 161 is shown the valve setting of one end of a large double-acting horizontal tandem producer-gas engine. The valve setting of a reversible two-cycle marine Diesel engine is shown in Fig. 162. In this engine the reversing angle—the angle through which the valve diagram must be displaced in order for the engine to run normally in the reverse direction—for the scavenging air is 58° − 26° or 32°, and for the fuel injection is 41° − 5° or 36°.

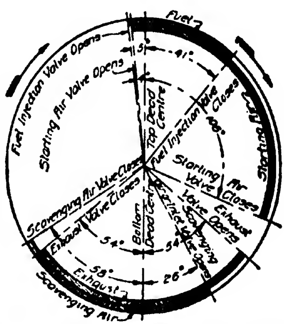


Fig. 162. Valve Setting Diagram for a Reversible Two-Cycle Marine Diesel Engine

Typical Valve Timing. The respective lags and leads of a high-speed motor—automobile, marine, or flying-machine motor—should be greater in proportion as the motor is intended to run at higher rotary speed. For a motor intended to run at normal speed, say a 4×5-in. motor to run at 1200 revolutions per minute, the following timing would be suitable:

Exhaust opens	40° ahead of bottom dead center
Exhaust closes	5° past top dead center
Inlet opens	10° past top dead center
Inlet closes	20° past bottom dead center

In motors intended to run at very high speed, and consequently provided with valves of very large diameters, the timing may be made as follows:

Exhaust opens	45° ahead of bottom dead center
Exhaust closes	10° past top dead center
Inlet opens	15° past top dead center
Inlet closes	30° past bottom dead center

If the high-speed motor is timed to give the very best output at high speeds it will not run so satisfactorily at low speed, and will not be as flexible. This is due to the fact that when the exhaust valve opens very early, some of the power otherwise available at low speed is lost through the exhaust, and when the inlet closes very late, some of the charge drawn in during the suction stroke will be forced out again during the beginning of the compression stroke.

The correct valve setting of a Knight sleeve-valve automobile motor is approximately as follows:

Exhaust opens	55° ahead of bottom dead center
Exhaust closes	15° past top dead center
Inlet opens	5° past top dead center
Inlet closes	40° past bottom dead center

STARTING

General Nature of Problem. A gas engine will not start itself in the way a steam engine does, when steam is turned on. It is necessary to get the engine in motion by means of some special source of power, before it can take up its normal cycle of operations. Generally, this special source of power is not adequate to get the engine moving rapidly when it is connected to any considerable load; it is always preferable, and often necessary, to throw the load completely off the engine till it gets under way.

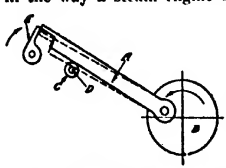


Fig. 163. Ignition-Retarding Device

In the normal running of an engine, the ignition of the charge occurs before the end of the back stroke; and, if the time of ignition is kept the same when starting, there is the possibility, often the certainty, that the high pressure of the explosion acting on the piston before the crank has reached dead center, will overcome the inertia of the engine, which is small because of its low speed, and will reverse its direction of rotation. The ignition has to be retarded by some device, so that it will not occur till after the crank has passed dead center. An example of a device for retarding the ignition (with make-and-break ignition) is given in Fig. 163. The igniter rod *A* (compare with *f*, Fig. 117), which is worked by a crank on the side shaft *B*, is supported during normal running on the reel *C*, which is loose on the fixed spindle *D*. In the position shown, it is just about to trip the interrupter lever *E* on the spindle carrying the movable electrode. When starting, the reel *C* is slid along the spindle *D* so that the igniter rod *A* rests, as shown in the dotted lines, directly on *D*; consequently the tripping occurs later.

Hand Starting. There are several general methods of starting gas engines. If the engine is small, not exceeding 10 horsepower, and can be disconnected from its load, it is common to start it by turning it over by hand for a few revolutions, till an explosive mixture is admitted and ignited. As it is difficult to pull the engine over when the charge is compressed for the whole back stroke, most engines are provided with an extra exhaust cam which is put into action while starting, and which not only opens the exhaust valve during the exhaust period, but also opens it again during the first part of the compression period, so that some of the explosive mixture is forced out of the cylinder and the amount of compression decreased. The explosion of this diminished charge after the crank has passed the dead point, starts the engine going; and after operation under these conditions for several cycles, the engine will come up to speed if it is not loaded heavily, and the compression and ignition may then be changed back to the normal running conditions.

Compressed-Air Starting. With large engines it is impracticable to start by hand, and other devices have to be used. One of the simplest and most certain is to start the engine by the admission of compressed air, which acts on the piston just as steam does in a steam engine. This method is especially desirable in an engine with

several cylinders, in which case one cylinder is used as a compressed-air cylinder to run the engine till the other cylinders take up their normal cycle of operations; and then the compressed air is shut off and the first cylinder is put into normal action. If the engine has only one cylinder, it may be brought to a good speed by the admission of compressed air; and then, after the compressed air is shut off, it will continue to revolve by its own inertia until an explosive mixture is taken in and exploded.

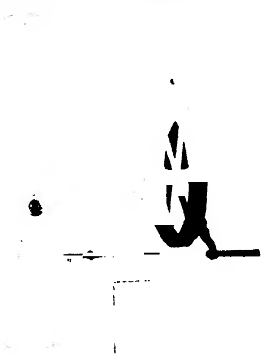


Fig. 164. Typical Air Compressor

An arrangement for starting a multicylinder engine with compressed air is illustrated in Figs. 49 and 164. A compressor, Fig. 164, which is driven by

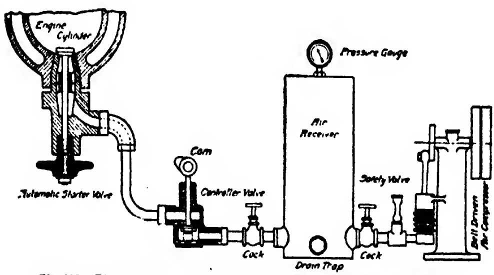


Fig. 165. Diagrammatic Arrangement of Cam-Controlled Compressed-Air Starter
Courtesy of Crossley Gas Engine Company, Manchester, England

a belt from the engine, forces air into a storage tank, and brings it to a pressure of about 160 pounds. In case of need the compressor can be operated by hand. When the engine is to be

started, the compressed air can be admitted to one of the cylinders. The cam B, Fig. 49, on the upper shaft is first thrown out of action by a special device, so that the inlet valve J cannot open. The hand lever on the outside of the crankcase near the cam A is thrown over, putting the ordinary exhaust cam A out of action, but bringing into action a double cam which keeps the exhaust valve E open throughout every upstroke of the engine. Another cam on the same shaft is brought into action at the same time, and operates a starting valve on the pipe from the compressed-air reservoir,

Fig. 166. Starting Gear of Fairbanks-Morse Engine
Courtesy of Fairbanks, Morse, and Company, Chicago

admitting compressed air to the cylinder on every down stroke. The cylinder then acts as a compressed-air engine till the explosions begin in the other cylinders, when the cams B and A are brought back to their normal positions and the starting cylinder functions normally. In other engines, compressed air is admitted to the cylinder during the expansion stroke, by manual operation of a special valve. After two or three admissions during successive cycles, the engine will attain speed enough to permit the opening of the gas valve and the commencement of the cycle.

The arrangement in Fig. 165 shows diagrammatically a belt-driven compressor and the other accessories. The air is admitted from the receiver at the proper time through the action of the cam-controlled valve.

Combined Hand- and Ignition-Starting. With engines up to 50 horsepower, a common method of starting is to ignite a charge which has been drawn into the engine by turning it over by hand. The engine is brought to the beginning of the expansion stroke, and a definite amount of gasoline is put into a cup which connects with the cylinder through an open cock. The engine is then pulled over till the piston has made half its forward stroke, air being drawn in and forming an explosive mixture with the gasoline which enters at the same time. The gasoline valve is then closed, and the engine turned quickly in the opposite direction; the charge is compressed as much as possible, and then ignited. The ignition is brought about by tripping the electric igniter by hand, or by the use of a special detonator. It is not possible, with a loaded engine, to compress the charge much by hand, so that this method is applicable only to engines of moderate size which can be disconnected from their starting load.

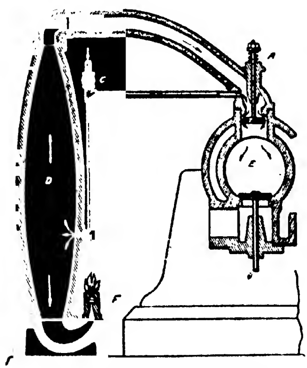


Fig. 167. Method of Supplying Engine with Compressed Charge

If the engine has to start under moderate load, it is generally necessary to supply it with a charge which has been compressed to a high pressure. This can be accomplished by setting the engine with the crank about ten degrees past the dead center on the expansion stroke, and then pumping an explosive mixture into the cylinder, Fig. 166, till the piston begins to move. At that instant the charge is ignited, and the work done by the expansion of the exploded charge

will be enough to start the engine on its cycle of operations. Another method of accomplishing the same thing is to connect the cylinder *E*, Fig. 167, with a special starting chamber *D*. When the engine is being shut down, the special inlet valve *A* is lifted from its seat, so that at each suction stroke air is drawn through the chamber *D* by way of the valve *F*. The chamber *D*, the cylinder, and the connecting pipe are thus filled with pure air at atmospheric pressure. When the engine is to be started, the gas cock *C* is opened and gas flows both into the chamber *D* and into the cylinder, a cock on the cylinder being opened. A pilot light burns across the opening above the valve *F*, and after a short time a combustible mixture of air and gas issues and catches fire. If the cock *C* is then closed, the flow of the explosive mixture stops, and the flame consequently shoots back past the valve *F* and ignites the mixture in *D*, closing the valve *F* against an upper face by the force of the explosion. The flame proceeds to the cylinder, the contents of which will have been compressed by the explosion in *D*, and causes an explosion there.

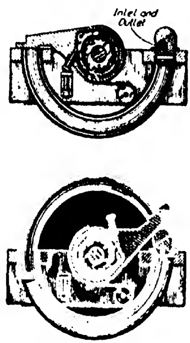


Fig. 168. Crescent Air Crank for Marine or Automobile Motor—Top View Showing Normal Position of Crank; Lower View Showing Action of Device in Cranking Motor

Courtesy of Gray Motor Company, Detroit, Michigan

Starting Automobile and Marine Motors. *Electric Motor.* Automobile and marine motors are often started either by an electric motor or by compressed air. The starting electric motor has a pinion on its shaft which can be engaged and disengaged with gear teeth cut in the flywheel rim. The motor is supplied with current from a storage battery, which is recharged by a generator which is always in gear with the engine. This generator is also used to supply current for the electric lights on the car or boat. The starting switch is operated by the pedal that moves the motor drive pinion in mesh, so that as the gear meshes the electric motor is started and when the gasoline engine begins to function normally the pedal is returned to its original position, thus disengaging the drive pinion and stopping the motor.

Compressed Air. If compressed air is used, the usual arrangement is to lead the compressed air from the storage tank to a rotary distributor, driven by the camshaft at half the crankshaft speed, which distributes the air at the proper time to pipes leading to each cylinder, the pipe being connected to the cylinder through a check valve which prevents the explosion blowing back into the starting air pipe.

Air Crank. Another device for starting automobile or marine motors is known as an air crank. As shown in Fig. 168, this apparatus consists of a frame, a short shaft and ratchet, and a crescent-shaped cylinder, in which travels a piston and curved crank arm that carries a pawl. This air crank is mounted in front of the flywheel, and its shaft is attached to the front end of the engine shaft by a flexible joint.

When the operator touches a push button, a charge of air throws the crank arm over half a revolution, the pawl engages the ratchet, and the motor is cranked in the natural manner, only with such speed and power as to give the motor several complete revolutions, a quick suction on the carbureter, and a hot spark from the magneto. In most cases, one or two throws are sufficient, but, if necessary, the engine can be cranked about fifty times, as rapidly as the button can be pushed.

Starting Large Engines with Compressed Air. Large gas engines are generally started by compressed air. One system is the same as that described for automobile and motor-boat engines. In another system, each cylinder end is fitted with a double check valve—one spring-loaded valve is held against its seat by the explosion pressure in the cylinder, the other check, or "pilot valve", as it is called, is held against its seat whenever the starting air is turned on to the system. This pilot valve can be opened by a rod, which is actuated by a cam on the lay shaft and, when open, admits the air to the first valve mentioned—the cylinder check—which is opened by the pressure of the compressed air, thus admitting the air to the cylinder. In this way each cylinder end in turn receives an impulse from the compressed air during its normal expansion stroke, the compressed air is exhausted during the normal exhaust stroke, and suction and compression of the mixture occur as in the normal running, to be followed by explosion. As soon as all cylinder ends are firing regu-

larly, the air is shut off, and the pilot valves drop by gravity allowing the push rods to drop far enough so that they are not actuated by the cams. In this way the starting mechanism is entirely out of gear except when it is being used for starting.

With any compressed-air starting gear in which all cylinders are both air cylinders and combustion cylinders during starting, great care must be used not to open the gas throttle too wide in starting, since, if too much gas is present with the high-pressure air, the explosion pressure may be excessive and enough to wreck the engine.

COOLING

Air Cooling. In very small engines, it is possible, when the engine is placed in a strong current of air, to replace the water jacket by a system of thin metal ribs (Fig. 169) or points on the external surface of the cylinder. The current of air can be obtained either from a fan driven by the engine, or, as in motorcycle or revolving-cylinder motors, by the movement of the engine itself. A cooling system in which a suction fan creates a current of air about each cylinder of a multicylinder engine is shown in Fig. 77 and described on page 135.

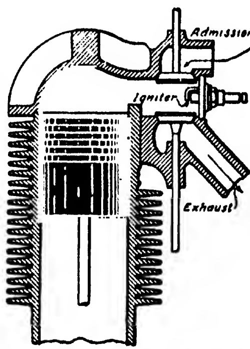


Fig. 169. Section of Cylinder Showing Fins Used for Air Cooling

Water Cooling. *Cylinder Barrel and Cylinder Head.* In all the preceding sectional views of gas-engine cylinders, it will be seen that the cylinder barrel and the cylinder head have double walls, and in every case provision is made for the active circulation of water through the space between the two walls. Without the use of a *water jacket* or some equivalent device, the engine would be inoperative, not only because the high temperature to which the cylinder would be raised by the explosions would vaporize the lubricating oil and cause the rapid destruction of the cylinder, but also because the entering mixture would be exploded before its time, by contact with

the hot metal. In large gas engines the cylinder and cylinder-head jackets are usually separate and are provided with independent water circulation systems.

Exhaust Valve. The necessity for effective cooling is greater in the larger engines; it is necessary to water-jacket the exhaust valve in large engines, in order that it may not be warped out of shape by the high temperature, and may not be hot enough to ignite the entering charge.

As can be seen from the sectional views of engines in earlier pages, the effective cooling of the exhaust-valve head is insured by conducting the inlet jacket water to the head by a pipe within the valve stem. The water overflows from the top of this pipe, cooling the head, and passes down between the pipe and the walls of the valve stem, thus also cooling the valve stem. Water is admitted and discharged from moving valves in a variety of ways: by means of flexible connections made of rubber hose; by swing-joint pipes; by telescope tubes; by a pipe sliding through a stuffing box into a chamber—a so-called "shot gun"; and the discharge may also be taken off by simply allowing the moving end of a pipe to discharge freely into a catch basin.

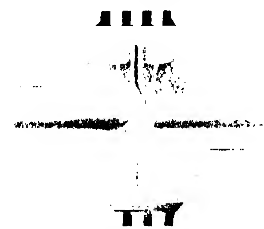


Fig. 170. Section of C. & G. Cooper Gas Engine Piston and Rod
Courtesy of C. & G. Cooper Company, Mount Vernon, Ohio

Piston and Piston Rod. Large single-acting and all double-acting engines must have their pistons water-cooled in order to prevent over-heating of the machine parts, pre-ignitions, lubricating and packing troubles. The jacket water can be admitted to and discharged from the moving piston in any of the ways enumerated above in the case of exhaust valves. The means for admitting and discharging the jacket water from a large trunk piston is shown in Fig. 63, page 119, and Fig. 100, page 163, and from a large single-acting piston fitted with a piston rod and crosshead, Fig. 102. In double-acting engines the piston rod must be cooled as well as the piston itself. This is an advantage, since it would be difficult to get the

water into and out of the piston if the piston rod had to be uncooled. The usual manner of cooling the rod and at the same time getting the water into the piston and out of it, is shown in Fig. 170. The rod is bored from both ends, leaving a partition midway of its length. The rod is then drilled, top and bottom, into the chambers thus formed; and the piston, which is provided with pipes in the jacket space to carry the inlet water to the bottom of the piston and to take the discharge from the top—thus insuring the cooling of the bottom of the piston and that the piston shall always remain full of water—is

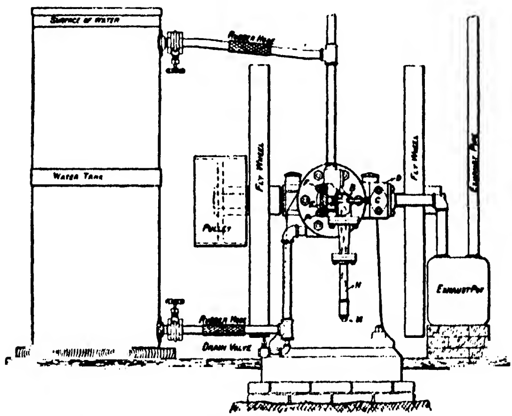


Fig. 171. Arrangement of Water Circulation for Engine Cylinder Jackets

forced on to the rod and up against a shoulder. In this way the water is introduced at one end of the rod and discharged from the other.

Industrial gases and blast-furnace, producer, or coke-oven gas often contain some sulphur, and if the piston rods are allowed to run too cold they will sweat and the moisture thus formed will combine with the sulphur in the gas to form sulphuric acid, which will attack the rod and other parts of the engine. In addition, if the rod sweats, it is impossible to properly lubricate it, which gives rise to packing troubles; and the fact that a film of oil is not protecting the rod makes

it more liable to attack from the sulphuric acid. To overcome these troubles the piston rod must be run warm enough to prevent it sweating. This is usually accomplished by first running the piston jacket water through the cylinder heads and then through the pistons, thus warming the water before it is introduced to the piston.

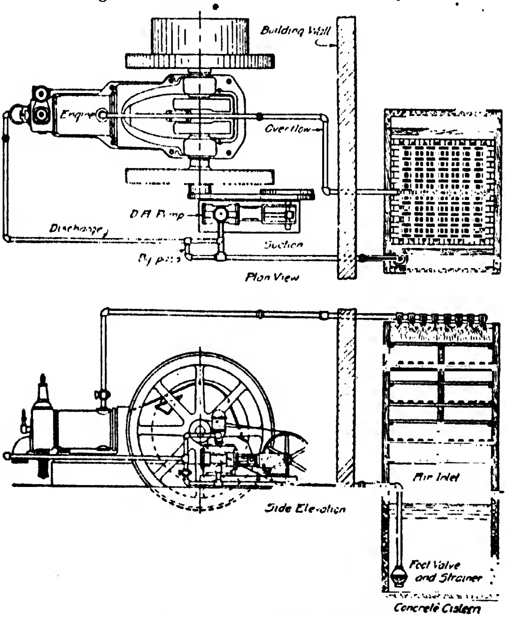


Fig. 172 Cooling Towers and Circulating System for Saving Jacket Water
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

Using Cooling Water Over Again. When the engine is water-jacketed, it is often practicable, with small engines, to use the same cooling water over and over again, and there is a distinct economy in so doing when the water must be paid for. The usual arrangement, Fig. 171, consists of a vertical galvanized-iron water tank of

considerable capacity, connected at its bottom to the lower part of the jacket, and near its top to the upper part of the jacket. The water in the jacket, being heated, rises and flows to the upper part of the tank, where it cools by contact with the air and with the sides of the tank. Cold water from the bottom of the tank flows to the cylinder jacket to take its place. A continuous circulation is maintained by the difference of density between the cold and the heated water. In large engines, when a large amount of water must be circulated, this method is generally too cumbrous; and the water is taken from some constant source of supply, such as the city mains, although if the location of the plant is such that water is expensive, a cooling tower or pond, in either of which the water is sprayed and cooled by contact with the air, can be, and generally is, installed.

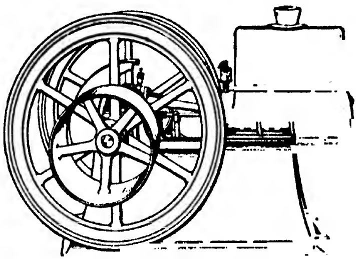


Fig. 173. Open-Jacket Gas Engine

A small, inexpensive installation of this type is shown in Fig. 172. The piping and valves are always so arranged that it is possible to draw the water from the jacket.

With portable engines, such as are used for agricultural purposes, it is generally impracticable to arrange for a water circulation in the jacket. In such cases it is usual to have a large water chamber on top of the cylinder, Fig. 173, communicating with the jacket and open on top to the atmosphere. The water in the jacket will gradually boil away, but may be replenished occasionally by pouring in a bucketful of water.

REGULATING FUEL MIXTURE

Air Supply. The air used in the engin may be taken from the engine room or from the outside. The inrush of air to the air pipe

makes a noise which is often objectionable in the engine room, but which can be greatly reduced if the air is taken from a large chamber, as in Fig. 28, where it is taken from the base of the engine.

Gas Supply. If the gas is taken from the city mains, the intermittent action of the engine in admitting gas will cause considerable fluctuation of pressure in the supply pipe, which is undesirable, because it makes variable the amount of gas admitted, and also causes flickering of any lights supplied from the same pipe. To reduce this fluctuation, it is usual to insert in the gas supply pipe a rubber bag which partly collapses during the admission stroke and fills out again during the other strokes. Any enlargement in the gas supply pipe will serve the same purpose, but the flexible rubber bag is more effective than a mere enlargement.

Methods of Mixing. The air and gas should be mixed as thoroughly as possible on their way to the cylinder. This is satisfactorily accomplished if the air and gas have to pass through a common admission valve after they are mixed, as in Figs. 49 to 67.

Regulating Strength of Mixture. The method of varying the strength of the mixture depends upon whether the gas is rich or lean and whether it is supplied to the engine under pressure, or not. For instance, if illuminating gas is used, which is supplied under pressure, the strength of the mixture is adjusted by throttling the gas supply, the air supply being left uncontrolled. On the other hand, if the engine is run on producer gas made in a suction producer, for which the engine furnishes the suction, the air supply alone is throttled to vary the strength of the mixture, since by closing off on the air supply the suction on the producer is increased, the amount of gas sucked into the engine in proportion to the amount of air is increased, and thereby the strength of the mixture is increased. These are the two extreme cases and for any intermediate case a combination of both methods of control is adopted; both the gas and the air supply are throttled in the correct proportion to get the best possible mixture.

EXHAUST

Mufflers or Silencers. If allowed to escape direct from an exhaust pipe of uniform cross-section, the exhaust is a source of annoyance by reason of the loud noise which it makes.

As the expanded charge is generally at a pressure of thirty to

fifty pounds above atmospheric pressure at the moment of the opening of the exhaust, the exhaust starts with very great velocity; and if permitted to go directly to the air, it makes a detonating noise.

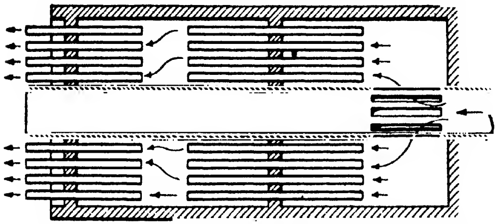


Fig. 174. Slits in Exhaust Pipe and Pipes in Muffler for Effecting Silent Exhaust

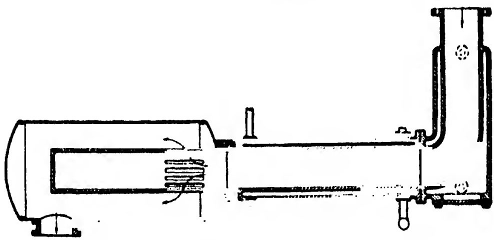


Fig. 175. Water-Jacketing Exhaust Pipe with Slits Opening into Muffler for Reducing Noise of Exhaust

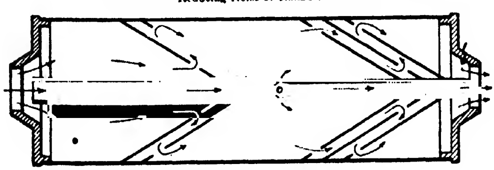


Fig. 176. Method of Silencing Exhaust by Passing Gases through Holes in Sheet-Metal Cones Inside Muffler

To reduce or prevent this noise, various devices, known as mufflers or silencers, are in use. For a silent exhaust, the gases should escape at a comparatively slow rate; and this has to be accomplished by lowering their pressure.

TABLE XV

Comparative Heat Losses in Large Gas and Diesel Engines

	Large Gas Engines B.t.u. per h.p.		Diesel Engines B.t.u. per h.p.	
Hourly heat consumption.......	11100—9150		7950—7350	
Hourly heat lost to the jacket water.		2800—2650		2000—1800
Hourly heat lost in the exhaust gases		4350—3300		2550—2200
Total hourly heat wasted........		7150—5950		4550—4000

The simplest device is to have the exhaust pipe discharge into an exhaust chamber or pot, as in Fig. 171, before going to the air; the enlargement of volume causes a corresponding fall in pressure, and the final escape to the air is consequently at a diminished velocity. It is better, in larger engines, to have the admission of the gases to the silencer through slits in the exhaust pipe, Figs. 174 and 175, as this prevents too sudden an expansion into the exhaust chamber. The escape of the gases from the muffler may be either through numerous holes in the periphery of the muffler; or through short lengths of small-sized pipe, Fig. 174; or through holes in sheet-metal cones fixed inside the muffler, Fig. 176. The repetition of the muffling device, forcing the gases to go through holes in concentric drums; or through two sets of pipes, Fig. 174; or through several cones, Fig. 176; insures quieter action of the exhaust, but at the expense of some back pressure in the cylinder, which, in bad cases, may seriously decrease the effective work done there. In Fig. 176 a central tube is shown, to which the gases have free access, and from the end of which they escape with high velocity; this gives a so-called *ejector* action to the muffler, the high velocity of the gases escaping from the central tube creating a partial vacuum in the nozzle and helping to suck the gases through the muffler.

It is claimed that an increase of power can be obtained with a well-designed muffler using this principle. In large engines it is customary to jacket the exhaust pipe, Fig. 175, and also to inject some of the jacket water into the pipe. This has the effect of lowering the pressure of the exhaust gases by cooling them; and it is a most

satisfactory method, since it brings no back pressure upon the engine. It is not, however, sufficient by itself to make the exhaust noiseless.

For automobile use, the muffler is usually made about twelve times the volume of a single cylinder, however many cylinders there may be. It is also provided in some cases with a *muffler cut-out*, permitting direct exhaust to the atmosphere, which is useful in case of the muffler becoming clogged, and also for ascertaining, by the noise of the exhaust, whether the engine is exploding regularly. To make the noise nearly imperceptible, a good plan in stationary practice is to have the pipe discharge near the bottom of a pit filled with large stones.

Utilization of Waste Heat. The major portion of waste heat in an internal-combustion engine is lost to the jacket water and in the exhaust gases, and amounts to about two-thirds of the total heat consumption. In the case of large installations the heat is distributed approximately as shown in Table XV. Where warm water of 100 to 150° F. can be used in the plant—for heating the building, for instance—the jacket water can be used directly.

The exhaust gases themselves are not adapted to the development of power and, therefore, their heat must be absorbed as close to the engine as possible by a better heat medium such as the hot jacket water. This is done in a cast- or wrought-iron heater. In the transmission about 40 to 70 per cent of the exhaust heat is absorbed, so that in large gas engines about 2000 B.t.u. and in Diesel engines about 1400 B.t.u. per horsepower hour of the engine rating is recovered. In the design of the heater, care must be taken that the back pressure on the engine is not increased seriously; the cooling, and consequent contraction, of the gases tend to diminish the resistance. In any case the back pressure will be greater than with a free exhaust.

In a 1350-horsepower blast-furnace-gas engine, equipped with an exhaust heater, 2405 pounds of jacket water per hour have been evaporated into steam at 100 pounds gage pressure, representing 30 per cent of the waste heat in the exhaust gases and jacket water available in the form of steam, or 20 per cent of the heat delivered to the engine. If this steam were used in a turbine about 20 per cent of it, or 4 per cent of the heat delivered to the gas engine, would be converted into work, thus increasing the thermal efficiency of the plant 4 per cent.

TABLE XVI

Heat Cost of Various-Priced Fuels

Fuel	Representative Prices	B.T.U. for One Cent
Acetylene, from carbide	$0.10 per lb.	500
Denatured alcohol	.40 per gal.	2,000
Water gas	1.00 per 1000 cu. ft.	3,000
Air gas, from gasoline	.25 per gal.	6,000
Coal gas	1.00 per 1000 cu. ft.	6,500
Gasoline	.20 per gal.	9,000
Kerosene	.15 per gal.	12,500
Natural gas	.50 per 1000 cu. ft.	18,000
Charcoal	.10 per bu. (15 lb.)	20,000
Petroleum	.05 per gal.	30,000
Producer gas, from anthracite	7.00 per ton	30,000
Producer gas, from coke	5.00 per ton	36,000
Anthracite	7.00 per ton	46,000
Producer gas, from soft coal	3.00 per ton	50,000
Coke	5.00 per ton	54,000
Mond producer gas, from soft coal	3.00 per ton	65,000
Soft coal	3.00 per ton	80,000

COST OF FUEL

Relative Cost of Different Fuels. Cost is one of the most important factors determining the choice of fuel in any engine. In Table XVI is given the number of B.t.u. that can be bought for one cent with fuels at the stated prices.

The relative cost of power developed by oil, gas, and steam engines depends on the cost of the oil and of coal, and this varies with the locality and the kind of oil or coal. In refining Pennsylvania or Ohio petroleum, not over ten per cent of the oil can be collected as gasoline, so that this oil, which is the easiest to use, is not available in as large quantity as the heavier oils, and, consequently, has a considerably higher cost. Kerosene forms twenty-five to fifty per cent of the crude oil and is consequently cheaper. Fuel oil and crude oil are the cheapest, but are also the most difficult to burn satisfactorily.

DESIGN DATA

The figures given in the following pages represent average American practice and, in most instances, European practice as well.

Usual Compression Pressures. In Table XVII are given the range of compression pressures and the average practice for the most common fuels as used in the various types of engines.

TABLE XVII

Usual Compression Pressures for Different Fuels and Engines

Fuel	Type of Engine	Compression: Above Atmospheric Pressure, Lb per sq in	Compression: Average Practice, Lb. per sq in (gage)
Gasoline in carbureter	Automobile	45—95	65
Gasoline	Stationary	60—105	70
Kerosene	Hot bulb, 250–500 r.p.m.	30—75	60
Kerosene	Vaporized before entering cylinder	45*—85†	65
Alcohol	Vaporized before entering cylinder	120—210	150
Fuel oil	Injected into hot bulb before compression—Hornsby-Akroyd	45	45
Fuel oil	Injected after compression, Franchetti-Otto Cycle	255	255
	Diesel Cycle	510	510
Natural gas 35–92%—0.0% H_2	Medium and large engines	75—160	120
Coke-oven gas	Large engines (in Germany)	105—135	120
Coal gas	Mostly small; very few large engines	75—120	100
Carbureted water gas	Mostly small; very few large engines	75—105	90
Producer gas 18%—5% H_2	Both large and small engines	100—160	130
Blast-furnace gas	Largest engines built	120—190	155

* With hot mixture without water injection.

† With water injection.

Compression Spaces. The compression space, expressed as a percentage of the piston displacement, for four-cycle engines (assuming the absolute suction pressure as 12.8 pounds per square inch) is, on the average, as in Table XVIII.

TABLE XVIII

Per Cent of Clearance for Various Types of Engines

Types of Engines	Compression Pressure Above Atmospheric Lb. per sq. in.	Clearance Per Cent
Gasoline engines, etc.	42.5	about 40
Illuminating-gas engines	85	about 25
Producer-gas engines	130	about 15
Blast-furnace-gas engines	155	about 12
Diesel oil engines	500	7–8

TABLE XIX

Average Mean Effective Pressures Obtained with the Various Fuels

Fuel	Mean Effective Pressure Lb. per sq. in.	Relative Capacity Referred to Illuminating Gas Engine as Unity
Natural gas	85.0	1.10
Illuminating gas	77.5	1.00
Coke-oven gas	77.5	1.00
Producer gas	62.5	0.81
Blast-furnace gas	57.5	0.74
Gasoline	75.0	0.97
Kerosene	55.0	0.71
Alcohol	55.0	0.71
Crude oil (Diesel)	100.0	1.29

Mean Effective Pressures. Table XIX gives the mean effective pressures obtained with various fuels, and also the relative capacity of the different types of engines, referred to an illuminating-gas engine as the standard.

Diagram Factors. The real value of the maximum explosion pressure or temperature may be found approximately by multiplying the maximum pressure or temperature, obtained from the ideal diagram—adiabatic compression and expansion—by a reduction factor which takes into account the decrease of temperature or pressure due to heat losses, cooling, etc. The value of this factor is not far from that of the card factor, or ratio between the real and ideal efficiencies. Table XX gives the values of these factors as found from practice for the various fuels.

Diesel engines operating at rated load with fuel injection for about 10 per cent of the stroke have a diagram factor of from 50 to 70 per cent. The diagram factors of two-cycle engines may be taken as 0.8 of those for four-cycle engines operating on the same fuel.

Actual Exponents of Compression and Expansion Curves. *Exponent of the Compression Line, from Practice.* The value of the polytropic exponent n is not a constant for the entire compression line, owing to the varying heat interchanges between the gases and the walls during the compression. The mean value varies from 1.30 to 1.38 for ordinary types of engines, with an average value throughout the compression of about 1.35. Imperfect cooling or high wall

TABLE XX

Diagram Factors for Explosion Engines

Fuel	Compression Lb. per Sq. In.	Diagram Factor Per Cent
Natural gas	90 – 140	40 – 52
Illuminating gas	80	45
Coke-oven gas	100 – 135	45
Producer gas	100 – 160	40 – 56
Blast-furnace gas	130 – 180	30 – 48
Gasoline	80 – 105	40 – 65
Kerosene vaporized before suction	45 – 75	30 – 40
Kerosene injected	may run as low	as 20
Alcohol	75 – 210	72 – 74

temperatures may raise the value of n above the adiabatic value 1.405, while loss of charge, because of leaky piston-packing or valves, will give apparent values of n that are too small.

Exponent of the Expansion Line, from Practice. The polytropic exponent n of the expansion line varies throughout the expansion. At the beginning of expansion, when the cooling surface of the cylinder is small and losses to the jacket are more or less balanced by continued burning of the charge, the value of the exponent, in most cases, approaches the adiabatic value. In some cases the heat from continued burning may overbalance the entire heat loss, in which event the value of n approaches unity. The greater the amount of cooling surface uncovered by the piston, the greater are the heat losses and the greater the corresponding values of n; but, near the outer end of the stroke, the rise in the value of n is less rapid, because of the rapid fall in temperature of the gas and the consequent smaller loss of heat to the jacket. Frequently, the value of n varies irregularly, alternately increasing and decreasing, obeying no definite law. An increase of piston speed, by reducing the time for heat losses and consequently the amount of the losses, decreases the mean value of n for the entire stroke. From the above, it is apparent that an accurate average is very difficult. The mean value of n varies generally between 1.30 and 1.50, although sometimes an indicator card shows as high a value as n equals 1.70. A loss of the charge through leaks at various points increases appreciably the apparent value of the exponent, while a sticky indicator gives an apparent error in the opposite direction.

TABLE XXI

Allowable Piston Speeds for the Various Sizes and Types of Engines

Engine		Allowable Piston Speed (S)	
Horsepower	Type	Limits	Average Practice
Above 1000	Stationary	700—1000	850
1000	Stationary	700—1000	800
700	Stationary	700—900	750
500	Stationary	650—850	700
150	Stationary	600—800	650
50	Stationary	500—700	600
Small	Stationary	450—700	550
	Automobile	600—1400	750

Allowable Piston Speed. The allowable limits of piston speed and average practice for different types of engine of sizes varying from small to over 1000 horsepower, are given in Table XXI.

Allowable Gas Velocity. The allowable mean gas velocity through the valves is given by Güldner as 4500 feet per minute. It is sometimes impossible, however, to realize mean velocities as low as 4500; in fact, in large engines the velocity is as high as 6000 to 8500 feet per minute. With a mean velocity through the valves of 4500 feet per minute the maximum velocity is approximately 7200 feet per minute.

It is good practice to make the diameter of the exhaust-valve seat larger than that of the inlet valve in order to get as much as possible of the products of combustion out of the cylinder before dead center is reached—the greater diameter giving a larger area with the small lift that the valve has before dead center. The exhaust pipe, from the cylinder to the muffler, should have a cross-sectional area of from 1.1 to 1.3 times the area of the free cross-section of the exhaust valve.

The allowable velocity in the air- and gas-inlet pipes and passages may be taken as from 1800 to 3600 feet per minute, depending upon the length of the line—the higher figure holding for lines up to 35 feet in length.

Volumetric Efficiency. An ideal engine would take into the cylinder, during each admission period, a charge equal in volume to the piston displacement and which would be at the atmospheric pressure and temperature. The ratio of the actual

TABLE XXII

Influence of the Height Above Sea Level on the Volumetric Efficiency

Height above Sea Level Ft. . . .	0	500	1000	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000	8000	10000
Atmospheric Pressure Ave. Barometer Reading (b) in. of Hg. . .	29.92	29.41	28.85	28.33	27.82	27.31	26.82	26.35	25.85	25.37	24.92	24.46	24.00	22.17	20.34
Pounds per square inch	14.70	14.35	14.17	13.91	13.67	13.41	13.18	12.93	12.71	12.46	12.23	12.03	11.78	10.88	9.98
Relative Suction Efficiency (b ÷ 29.92).	1.000	0.984	0.965	0.948	0.931	0.914	0.892	0.862	0.865	0.848	0.833	0.818	0.803	0.742	0.681
Loss of Suction Efficiency due to height—in per cent	0.0	1.6	3.5	5.2	6.9	8.6	10.8	11.8	11.5	15.2	16.7	18.2	19.7	25.8	31.0

weight of charge admitted to the weight of the ideal charge is called the volumetric efficiency. In order to obtain as large an amount of work per stroke as possible, and thus as favorable a use of the cylinder volume as is attainable, the volumetric efficiency must be made as large as possible by diminishing the suction and exhaust resistances and the suction temperature—in other words, there must be as nearly complete cylinder filling as can be obtained. Since the absolute pressure of the suction stroke equals the barometric pressure minus the suction of the charging stroke, the barometric pressure also exerts considerable influence on the volumetric suction efficiency—and through that on the mean effective pressure and the indicated and brake horsepowers.

Influence of Altitude on Volumetric Efficiency. The decrease in volumetric efficiency must be especially reckoned with in the case of plants located at a very high altitude, in consequence of the diminished barometric pressure. Because of the altitude the density of the charge, the volumetric efficiency of the suction stroke, and finally, the power of the engine, is decreased.

Table XXII shows the effect of the diminished barometric pressure, due to an increase in altitude, upon the volumetric suction efficiency.*

Values of Absolute Exhaust Pressure and Temperature. The common actual values of the absolute exhaust pressure p_r and temperature (T_r, absolute, and t_r degrees F.) are as follows:

$$p_r = 15.4 \text{ to } 16.4 \text{ lb. per sq. in., absolute}$$
$$T_r = 1260° \text{ to } 1440° \text{ F., absolute}$$
$$t_r = 800° \text{ to } 980° \text{ F,}$$

Too early closing of the exhaust valve, and very long or too small exhaust pipes, may increase these values considerably.

Common Values of Absolute Suction Pressure p_s, and Volumetric Efficiency E_v. *For Slow-Speed Engines with Mechanically Operated Inlet Valves,*

p_s = 12.5 to 13.5 lb. per sq. in., absolute
= 4.5 to 2.5 inches of mercury suction
E_v = 0.87 to 0.90

For High-Speed Engines with Mechanically Operated Inlet Valves,

p_s = 11.4 to 12.1 lb. per sq. in., absolute
= 6.8 to 5.3 inches of mercury suction
E_v = 0.78 to 0.83

For Slow-Speed Engines with Automatic Inlet Valves,

p_s = 12.1 to 12.8 lb. per sq. in., absolute
= 5.3 to 3.9 inches of mercury suction
E_v = 0.80 to 0.85

For High-Speed Engines with Automatic Inlet Valves,

p_s = 11.1 to 11.8 lb. per sq. in., absolute
= 7.4 to 6.0 inches of mercury suction
E_v = 0.65 to 0.75

For Very High-Speed Auto Engines with Automatic Inlet Valves and Air Cooling,

p_s = 8.5 to 10.7 lb. per sq. in., absolute
= 12.7 to 8.2 inches of mercury suction
E_v = 0.50 to 0.65

Suction producers and carbureters increase the suction resistance —in unfavorable cases they may decrease the above values of E_v by from 2 to 5 per cent.

The temperature T_s of the suction stroke equals 630° to 810° absolute; t_s equals 170° to 350° F.

Volume of Material for Foundations. The average volume of material in foundations for the various types of engines may be taken as follows:

For horizontal engines without outboard bearings

14-18 times the normal brake horsepower, in cubic feet

For horizontal engines with outboard bearings

21-25 times the normal brake horsepower, in cubic feet

For vertical engines without outboard bearings

7.7-8.8 times the normal brake horsepower, in cubic feet

For vertical engines with outboard bearings

9.8-10.5 times the normal brake horsepower, in cubic feet

Mechanical Efficiencies of Engines. In Table XXIII are given the average mechanical efficiencies found in practice for the various types of gas engines.

Relative Weights of Flywheels. The relative weights of flywheels required to give the same degree of regularity in speed of rotation with the same peripheral speed of the flywheel for different numbers and arrangements of cylinders and different fuels, are given in Tables XXIV and XXV.

Weights of Reciprocating Parts. Table XXVI gives the proper weights of the reciprocating parts per square inch of piston surface for explosion engines and constant-pressure engines of various types.

TABLE XXIII

Average Mechanical Efficiencies of the Various Types of Engines

Type of Engine	Mechanical Efficiency	
	4-Cycle	2-Cycle
Small, high-speed auto, multicylinder, single-acting	0.75	
Small, single-cylinder boat engine, single-acting	0.85	0.68
Small or medium, single-cylinder stationary, single-acting	0.87	0.7
Small or medium, two-cylinder stationary, single-acting	0.84	
Small or medium, three-cylinder stationary, single-acting	0.82	
Small or medium, four-cylinder stationary, single-acting	0.80	
Large, single-cylinder stationary, single-acting	0.90	0.70
Large, two-cylinder stationary, single-acting	0.86	to
Large, four-cylinder stationary, single-acting	0.84	0.80
Double-acting single-cylinder	0.83	0.75
Double-acting tandem, two-cylinder	0.81	0.73
Double-acting twin tandem, four-cylinder	0.77	0.69

TABLE XXIV

Comparison of Vertical Single-Acting 4-Cycle Engines Operating on Various Fuels

(Relative flywheel weights obtained from turning-moment diagram derived from actual indicator cards) (Güldner)

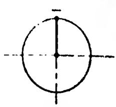
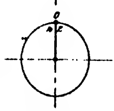
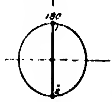
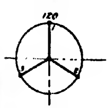
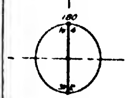
	Number of Cylinders	Crank Angle Degrees	Crank Travel Between Explosions, Degrees	Explosion Engines Operating on				Constant Pressure Oil Engines for Equal	
				Lean Gas (Producer, Blast-Furnace Gas, etc.) for Equal		Rich Gas (Illuminating, Coke-Oven Gas, Gasoline, etc.) for Equal			
				Cylinder Dimensions	Maximum Indicated Horsepower	Cylinder Dimensions	Maximum Indicated Horsepower	Cylinder Dimensions	Maximum Indicated Horsepower
I	1		720	1.00	1.00	1.00	1.00	1.00	1.00
II	2		360	0.86	0.43	0.85	0.425	0.89	0.445
III	2		540 & 180	1.44	0.72	1.20	0.60	1.17	0.585
IV	3		240	0.72	0.24	0.65	0.22	0.75	0.25
V	4		180	0.304	0.076	0.265	0.066	0.25	0.06

TABLE XXV

Comparison of Horizontal 4-Cycle Engines Operating on Lean Gases

(Relative flywheel weights obtained from turning-moment diagram derived from actual indicator cards) (Güldner)

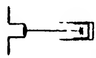
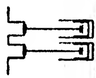
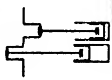
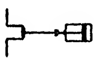
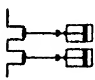
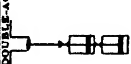
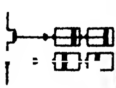
	Number of Cylinders	Cylinder and Crank Arrangement	Angle between Cranks Degrees	Crank Travel between Explosions Degrees	Relative Flywheel Weight for Equal	
					Cylinder Dimensions	Maximum Indicated Horsepower
I	1	SINGLE-ACTING 	...	720	1.00	1.00
II	2		0	360	0.85	0.425
III	2		180	540 & 180	1.20	0.60
IV	1	DOUBLE-ACTING 	...	540 & 180	1.20	0.60
V	2		0	180	0.62	0.155
VI	2_a		...	180	0.325	0.08
VII	4		90	90	0.28	0.035

TABLE XXVI

Weight of Reciprocating Parts for Various Types of Engines

Kind of Engine	Weight of Reciprocating Parts per Sq. In. of Piston Surface	
	Explosion Engines Lb.	Constant-Pressure Engines Lb.
Single-acting Engines:		
Trunk piston, short stroke ($l<1.5d$)	5.5— 8.5	7.0—10.0
Trunk piston, long stroke ($l\geq 1.5d$)	8.5—10.5	10.0—11.5
With crosshead ($l=1.5d$ to $1.33d$)	12.5—17.0	14.0—18.5
Two-cylinder tandem	17.5—21.5	19.0—23.0
High-speed automobile	0.35—0.60	
Double-acting Engines:		
Single-cylinder, without tail rod	14.0—18.0	
Single-cylinder, with tail rod	17.0—20.0	18.5—21.5
Two-cylinder tandem	21.0—25.5	23.0—27.0
Three-cylinder tandem (one of which is a blowing tub)	28.5	

NOTES:—l is length of stroke of the engine, d is diameter of the cylinder } both in the same units.
($l<1.5d$) is read, stroke less than 1.5 times the diameter.
($l\geq 1.5d$) is read, stroke greater than or equal to 1.5 times the diameter.

Thickness of Cylinder Walls. Table XXVII gives for cylinders of various diameters D the required thickness of cylinder walls t to resist the maximum explosion pressure, the reboring allowance x, and the total required thickness $t+x$. The required thickness to resist the maximum explosion pressure is based on a maximum explosion pressure of 350 pounds per square inch and an allowable tensile stress of 3500 pounds per square inch—which is high, but allowable since the maximum pressure acts only at the inner end of the cylinder barrel, which, in single-acting engines, is reinforced by a flange, and, in double-acting engines, by the flange connecting the cylinder barrel and jacket wall, and by the valve ports. Where a cylinder liner is not used a reboring allowance x should be added to this thickness. Where a liner is used, the reboring allowance x gives its necessary thickness.

Attendance. In spite of the belief that gas-power plants can be operated by cheaper men than steam plants, quite the reverse is true

TABLE XXVII

Required Thickness of Cylinder Walls to Resist Maximum Explosion Pressure for Cylinders of Different Diameters

$D=$	4″	6″	8″	10″	12″	14″	10″	18″	20″	24″	28″	32″	40″	48″
$x=$	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
$t=$	[illegible]	[illegible]	1″	[illegible]	[illegible]	[illegible]	1″	[illegible]	[illegible]	1[illegible]	1[illegible]	1[illegible]	1[illegible]	2[illegible]
$t+x=$	*[illegible]	*[illegible]	[illegible]	[illegible]	1[illegible]	1[illegible]	1[illegible]	1[illegible]	1[illegible]	1[illegible]	1[illegible]	2[illegible]	2[illegible]	3[illegible]

*To allow for the possible shifting of cores, etc., it is necessary that, for stationary engines, the thickness for one-piece cylinders up to about 8 inches in diameter should be at least [illegible] in., and for separate cylinder barrels at least [illegible] in.

if they are to be operated at a maximum of economy and with a minimum of trouble.

Güldner gives the following equations (based on practical experience) for the amount of attendance necessary in internal-combustion engine plants:

For illuminating, natural, coke-oven, and blast-furnace gas- and for oil-engine plants:

$$W = 0.25\ \sqrt{n \times \text{b.h.p.}}\ \text{hours}$$

For producer-gas plants:

$$W = 1.25\ \sqrt{n \times \text{b.h.p.}}\ \text{hours}$$

in which W is the time of attendance required in hours per day of ten hours, b.h.p. is the rated total capacity of the plant, and n is the number of units in the plant.

PERFORMANCE DATA

Fuel Consumption. *Rated Load.* The consumption of fuel in a gas engine running at its rated load, when natural gas is used, is from 10 to 15 cubic feet per b.h.p. per hour; with illuminating gas, 14 to 18 cubic feet per b.h.p. per hour; with producer gas, 75 to 110 cubic feet per b.h.p. per hour; with blast-furnace gas, 100 to 135 cubic feet per b.h.p. per hour—depending upon the size of the engine. These are average limiting results, large engines showing a higher economy than smaller engines.

The consumption of gasoline in an engine of small size averages about 0.1 gallon per b.h.p. per hour. In the Diesel motor, the average consumption of crude oil per b.h.p. per hour is considerably less than 0.1 gallon, the average being about 0.06 gallon.

TABLE XXVIII

Relative Increase of Fuel Consumption per Brake Horsepower per Hour at Partial Loads

	Full Load	¾ Load	½ Load	¼ Load
Natural-gas engine	1.00	1.05—1.20	1.20—1.50	1.75—2.00
Illuminating-gas engine	1.00	1.05—1.20	1.20—1 50	1.75—2.00
Producer-gas plant (Engine and producer)	1.00	1.15—1.25	1.45—1.60	2.30—2.70
Gas-producer	1.00	1 04—1.06	1.10—1.15	1.30—1.40
Blast-furnace-gas engine	1 00	1.05—1.20	1.20—1.50	1.75—2.00
Low-pressure oil engine	1.00	1 15—1 35	1.45—1.70	1 90—2 30
Diesel oil engine	1.00	1.02—1 15	1.07—1.25	1.40—1.90

Less Than Normal Load. The fuel consumption per horsepower hour increases perceptibly with a decrease of load below the normal, as is shown in Table XXVIII.

No-Load Consumption. In explosion engines the total no-load consumption goes as high as 30 to 45 per cent of the total consumption at normal load—in constant-pressure combustion engines this figure is only 20 to 25 per cent. In producer-gas plants the coal consumption increases faster at partial loads because of the loss of efficiency of the producer as well as of the heat consumption of the engine itself.

Fuel-Consumption Tests for Commercial Engines. Table XXIX gives a number of commercial test results for various engines, showing fuel consumption and mechanical efficiency.

Heat Losses at Various Speeds and Compressions. From Table XXX it is seen that the efficiency of an internal-combustion

TABLE XXX

Heat Losses at Various Speeds and with Various Compression Ratios (Meyer's Tests)

Volumetric Compression Ratio	R.P.M.	Mean Effective Pressure Lb. per Sq. In.	Ratio Gas to Air	Heat Value of Charge B.t.u.	Work Done by 1 B.t.u. Ft.-Lb.	Exhaust Temperature °F.	Heat Distribution Per Cent		
							Work	Jacket	Exhaust
2 67	187	54 3	7.11	18 5	140	1022	18 0	51 2	30.8
2 67	247	51 5	7 35	17 4	141	1137	18 1	45 6	36 3
4.32	187	69 3	7 43	17 0	190	867	24 4	53 8	21.8
4.32	247	65 2	7 40	16 8	184	992	23 7	49 5	26.8

TABLE XXIX

Fuel Consumption of Internal-Combustion Engines—Test Results (Abbreviations as in Table XXXI)

	Manufacturer	Fuel	Engine: Cycle	Type	S-A or D-A	Cylinder: No	Diam. Inches	Stroke Inches	Rated: B.H.P.	R.P.M.	Load Factor Per Cent	Lower Heat Value of Fuel B.t.u. per Lb.	Cu. Ft.	Fuel Consumption per B.H.P. per Hour Lb.	Cu. Ft.	Heat per B.H.P. Hr., B.t.u.	Mechanical Efficiency
A	Walrath	Natural Gas	4	V	S-A	3	13	14	75	250	106.0		1041		11.50	11,960	78.7
B	Westinghouse	Natural Gas	4	H T T	D-A	4	23½	33	1500	150	96.1		1041		10.0	10,430	78.8
											73.4		1064		10.5	11,000	75.4
											45.0		1064		12.0	12,600	68.4
											0.0				Per i.h.p.	15,830	0.0
	Koerting	Illuminating Gas	4	H	S-A	1				160			500		18.07	9,030	
	Westinghouse	Illuminating Gas	4	V	S-A	3				235			507.4		15.96	9,540	76.0
C	Westinghouse	Producer Gas	4	V	S-A	3	19	22	235	200	95.0		137.3		83.2	11,420	82.3
D	Wood	Producer Gas	4	H T	S-A	2	25	30	300	150	56.3		145.0		102.8	14,910	70.0
E	Westinghouse	Producer Gas	4	H T	D-A	2	23½	33	500	150	95.7		114.6		89.0	10,200	83.5
	Westinghouse	Blast-Furnace Gas	4	H T T	D-A	4	42	60	3000		68.6					13,175	
											106.0					10,300	
	Allis-Chalmers	Blast-Furnace Gas	4	H.T T	D-A	4	42	54	3000	83.1	93.5		98.3	Ave of yr '09	125.8	12,370	
											91.5	Ave of 1st 4 months of '10				12,100	
												Maximum monthly ave. '09				10,720	
												Maximum daily ave. '09				9,910	
	Westinghouse	Gasoline	4	V	S-A	2	5.75	6	10		85.8					14,710	80.7
	Daimler	Gasoline	4	V	S-A	4	3.56	5.11	16-20	400		17,300		0.875		15,300	86.1
										1000		17,300		0.753		13,170	79.8
F	De La Vergne, Type HA	Kerosene	4	H	S-A	1	8.2	14	*4.85	*213.6		18,000		1.06		19,580	81.6
G	De La Vergne, Type HA	Kerosene	4	H	S-A	1	14½	17	25	200	107.0	18,870		0.73		13,770	86.7
											71.8	18,870		0.92		17,390	80.2
											36.0	18,870		1.35		24,800	67.8
											0.0	18,870	Per i.h.p.	1.34	Per i.h.p.	25,240	0.0
	Diesel	Kerosene	4	V	S-A	1	15.75	23.65	70	160	99.5	18,610		0.43		7,990	79.1
	Snow-Diesel	California Crude Oil	4	H	S-A	1			110	200	100	17,500		0.429		7,506	
											87.5	17,500		0.417		7,295	
											75	17,500		0.428		7,490	
											50	17,500		0.466		8,155	
											25	17,500		0.580		10,150	
H	De La Vergne, Type FH	Crude Oil	4	H	S-A	1	17	27½	85	180	101.0	19,080		0.498		9,510	68.7
											76.1	19,080		0.485		9,270	66.6
	De La Vergne, Type FH	Fuel Oil, Solar	4	H	S-A				125	155	102.9	17,750		0.408		7,250	
I	Busch-Sulzer Bros. Diesel	Fuel Oil, Texas	4	V	S-A	3			225	164	106.9	18,990		0.444		8,430	
											97.6	18,990		0.441		8,375	
											72.4	18,990		0.454		8,670	
											49.5	18,990		0.482		9,150	
											22.1	18,990		0.780		14,810	
	Snow-Diesel	Fuel Oil, Solar	4	H	S-A				150	165	100	18,210		0.406		7,390	
											75	18,210		0.420		7,650	
											50	18,210		0.438		7,998	
	Snow-Diesel	Fuel Oil, Solar	2	H	S-A				450	150	100	18,300		0.440		8,007	
											75	18,300		0.450		8,300	
											50	18,300		0.560		10,180	
	Snow-Diesel	Mexican Crude Oil	2	H	S-A				450	150	100	17,900		0.48		8,600	
											50			0.49		8,570	
											25			0.53		9,275	

TABLE XXXI

Heat Balances of Gas and Oil Engines—Four-Cycle

	Manufacturer	Fuel	Engine		Cylinder			Rated		Load Factor Per Cent	Heat Balance, Per Cent					
			Type	Single or Double-Acting	No.	Diam. Inches	Stroke Inches	B.h.p.	R.p.m.		Indicated Horsepower	Brake Horsepower	Friction	Jacket	Exhaust	Per Cent Losses by Radiation and Unaccounted For
A	Walrath	Natural Gas.	V	S-A	3	13	14	75	250	105.0	27.1	21.3	5.8	49.5		23.4
B	Westinghouse	Natural Gas.	H.T.T	D-A	4	23½	33	1500	150	96.1	31.0	24.4	6.6	43.8		25.2
										73.4	30.6	23.1	7.5	45.7		23.7
										48.0	29.4	20.1	9.3	51.1		19.5
										0.0	16.1	0.0	16.1	56.3		27.6
C	Westinghouse	Producer Gas	V	S-A	3	19	22	235	200	95.0	27.1	22.3	4.8	30.4		42.5
D	Wood	Producer Gas	H.T.	S-A	2	25	30	300	150	56.3	24.4	17.1	7.3	25.0		50.6
E	Westinghouse	Producer Gas	H.T.	D-A	2	23½	33	500	150	95.7	29.9	24.95	4.95	34.2		35.9
F	De La Vergne Type HA	Kerosene	H	S-A	1	8.2	14	*4.95	*213.8		15.9	13.0	2.9	29.0		55.1
G	De La Vergne Type HA	Kerosene	H	S-A	1	14½	17	25	200	107.0	21.4	18.5	2.9	50.0		28.6
H	De La Vergne Type FH	Crude Oil	H	S-A	1	17	27½	85	180	101.0	38.4	26.8	11.6	30.2		31.4
										76.1	40.2	27.5	12.7	26.5	20.0	13.3
I	Busch-Sulzer Bros. Diesel Engine	Texas Fuel Oil	V	S-A	3			225	164	108.9		30.2		26.6		
										97.6		30.4		24.5		
										72.4		29.5		25.3		
										49.5		27.8		25.6		
										22.1		17.4		27.4		

*Actual

Abbreviations:

V	Vertical	S-A	Single-acting
H	Horizontal	D-A	Double-acting
H.T.	Horizontal Tandem		
H.T.T.	Horizontal Twin Tandem		

engine increases with an increase in the compression. It is further seen that an increase in the speed of rotation has no effect upon the total amount of heat lost to the jacket and in the exhaust, but only transfers part of the jacket loss to the exhaust; this being due to the fact that there is less time for the absorption of heat by the jacket.

An increase in the ratio of compression from 2.67 to 4.32 increases the heat loss to the jacket about three per cent, due to the higher temperatures obtaining in the cylinder resulting from the increased compression pressure with its corresponding increased explosion pressure, and decreases the heat loss to the exhaust about nine per cent, the net change in the two losses being the increase in the efficiency of the cycle.

Heat Balances of Four-Cycle Engines. Table XXXI shows the heat balances for 4-cycle gas and oil engines of various types with different fuels.

GAS-ENGINE OPERATION

General Classification of Troubles. For the successful operation of a gas engine, intelligent care and accurate adjustment are necessary, as well as an understanding of the processes going on in the cylinder. It sometimes happens that the engine fails to start, although the ordinary starting operations have been carried out faithfully. The most common causes of this difficulty are incorrect strength of mixture, failure of ignition, or leakage of the charge. The *setting of the gas valve* which gives a satisfactory mixture one day, may give a non-explosive mixture on the following day as a result of changes in the pressure or composition of the gas or other cause. The *strength of the mixture* should be varied in case of failure to start. If this is ineffective, the *ignition* should be tried. The batteries may have run down as a result of much use or of short-circuiting, and should be tested by short-circuiting momentarily, when they should give a bright spark. Too strong a current is undesirable, as it burns the contact points rapidly. It is well to have on hand a spare set of cells for putting in circuit. There should always be a switch in the battery circuit, which should be thrown out when the engine is shut down, so as to prevent short-circuiting. If the battery is in good condition, the trouble may be with the

TABLE XXXII

Schedule for Location of Gasoline Motor Troubles

The motor will not run	Ignition in working order		Good compression	Carbureter in working order		Ignition timing is incorrect Exhaust valve does not open Gasoline is too dense or contains water Gasoline tank is empty Gasoline feed-pipe is clogged or cock closed
				Carbureter not in working order		Carbureter flooded—needle valve needs grinding, float is too heavy, float leaks, or float or mechanism stuck Too little gasoline—needle valve setting bad, float is too light, feed-pipe or jet obstructed Auxiliary air-valve adjustment incorrect, or air valve does not seat, due to dirt on the valve seat or sticky valve stem Throttle valve loose on stem Balance lever broken
			Poor compression			Valve broken Valve stem sticks Valve spring broken or too weak Cylinder cracked or worn Broken piston rings Slots in piston rings are in line Piston rings gummed (inject gasoline) Leakage past valves (regrind them if corroded) Valve stems too long Valve timing incorrect Leakage past igniter (if make-and-break type) or spark plug not screwed in tight
	Ignition not in working order	With batteries	No spark at end of plug	Spark at trembler	Voltage or amperage sufficient	Trembler screws loose Trembler stuck Leak between coil and spark plug Broken or grounded secondary wire
					Voltage or amperage insufficient	Replace battery
				No spark at trembler		Batteries run down Leak in primary circuit Trembler dirty or too far from platinum point Timer misplaced
			Spark at end of spark plug (unscrew it)			Broken spark plug Dirty or moistened spark-plug Spark plug points broken or too far apart Voltage too low for the conditions inside the cylinder
		With magneto		Spark at the spark plug		Magneto out of time Wiring incorrect
				No spark at the spark plug		No contact (carelessness or broken wire) Make-and-break badly set, or damaged Carbon brushes worn, or used up Distributor contacts bad Wiring incorrect Grounded primary wire Water or moisture in parts Bad insulation of wipers or spark plugs
The motor starts, but stops after a few revolutions						Piston seizes from lack of lubrication or cooling water Too much oil in crankcase Water leaks into the cylinder from the jacket Poor carburetion or stuck float Exhaust clogged Water in gasoline tank Adjustment on auxiliary air valve too light

electrodes, through their having become fouled or wet; or, in the make-and-break system, through a gumming of the spindle of the moving electrode, which makes it sticky and slow in action. The igniter plug should be withdrawn, and the electrodes examined. The whole igniter circuit should be examined for short circuits.

If the trouble is not with the igniter, it may be caused by *leakage of the charge*. To test this, the engine, if not too large, is pulled over by hand. The resistance to turning, on the compression stroke, should be very considerable. If the resistance is not great enough, the compressed charge is escaping. The leakage may be either past the piston, the igniter plug, or the valves. If the leakage is past the piston, it is due either to the wearing of the cylinder or to the sticking of the piston rings. The latter is likely to occur after a while, especially if the cylinder has been permitted to get very hot; it can be remedied by taking the piston out and loosening and cleaning the rings with kerosene. A leakage past the valves is due either to gumming of the valves or to other deposit which keeps the valve off its seat, to wearing of the valve, or to sticking of the valve-stem in its guide as a result of imperfect lubrication. The gumming and wear of the exhaust valve are the most common causes of leakage, and may be remedied by grinding the valve on its seat with flour of emery and oil.

The presence of *water in the cylinder*, which has leaked in from the jacket through imperfect joints, sometimes causes the electrodes to become wet, and prevents the engine starting. In some engines the possibility of this particular trouble is avoided by a special design of the jacket in which there are no joints communicating with the inside of the cylinder.

Method of Locating Seat of Trouble by Use of Schedule. As the number of things which may occur in a gas engine to prevent its proper action is considerable, it is best to proceed systematically in hunting for the trouble when it arises. The most advantageous procedure to follow in any case, depends on the type of engine. An example is given in the schedule (Table XXXII and following) for a gasoline engine with jump-spark ignition, such as an ordinary automobile engine. If the motor refuses to operate, the first thing to do is to look to the gasoline supply. If that is all right, look to the ignition. By unscrewing the spark plugs, laying them on

the cylinders, and cranking the engine, it can be seen if the sparking is satisfactory. If it is satisfactory, try the compression. If that also is satisfactory, examine the carbureter; then see that the exhaust or inlet valves are operating properly. The method of following up the trouble, eliminating from consideration those things that are all right, is given in detail in the schedule. The right-hand column gives the actual causes of the observed troubles.

If make-and-break ignition is used, the procedure in investigating a failure of the ignition will naturally be different; it will also be much simpler.

It happens not infrequently that a gas engine will make a few revolutions, and will then stop. Some of the causes for this are also indicated in the schedule.

Investigating Quality of Mixture. If the motor runs fairly well and if the compression is all right, the spark plugs clean and properly set, magneto or batteries in good order, gasoline supply clean and other conditions satisfactory, the quality of the mixture can be determined in the following way with an automobile engine:

Not enough air or too much gasoline...	(1) Motor speeds up when air valve is pushed open slightly—turn low-speed nut down (2) Regular missing after pulling slowly on level, motor does not pick-up immediately if clutch is disengaged while running—mixture very rich in this case (3) After ascending long hill motor does not pick-up if clutch is disengaged (4) Motor will not pop back in carbureter on suddenly opening throttle (5) Motor picks-up well on opening throttle in just starting out
Too much air or not enough gasoline...	(1) Motor hard to start (2) Motor speeds up on closing auxiliary air valve—turn lower or low-speed adjusting nut, up a few notches (3) Upon opening throttle, car hesitates before picking-up speed (4) Tendency to pop back in carbureter when throttle is opened suddenly

Warm weather has the effect of not enough air. Cold weather has the effect of too much air.

Cylinder Oil. The cylinder oil that is commonly used in steam engines cannot be used in gas engines, as it carbonizes at the high temperature of the explosion, and forms a deposit in the cylinder and on the exhaust valve. A special oil is used; and even this, if

supplied in excess, causes a gradual accumulation of hard deposit in the cylinder, which must be cleaned out occasionally. Apart from its interference with the action of the igniter and exhaust valve, this deposit is liable to cause premature ignition by being raised to incandescence.

Stoppage of Jacket Water. Cold water must be kept circulating through the jackets whenever the engine is running, being started as soon as the cylinder warms up. A stoppage of this flow, even for a comparatively short time, is likely to have a disastrous effect upon the cylinder. A gradual accumulation of sediment may occur in the water jacket, with a consequent reduction in its efficiency. On shutting down, it is always better to drain the jacket, which not only prevents the possibility of its freezing up in winter, but also tends to clear it of sediment. In general practice, however, the jackets are drained only in cold weather.

Back-Firing. In the running of a gas engine—especially under light loads—very loud and alarming explosions are sometimes heard in the admission pipe or in the exhaust pipe. The back-firing in the admission pipe sometimes results from a leaky admission valve, at other times it is caused by something in the cylinder, such as a gas pocket, firing the entering charge before the inlet valve has a chance to close; the latter is the more frequent cause of the two. The explosions in the exhaust, indicating as they do the presence of explosive gases in the exhaust pipe, are caused either by the use of a mixture which is too weak or too rich, or by faulty ignition. If the mixture is too weak, the charge taken in just after an explosion may fail to ignite, because it is mixed with the products of the previous explosion, while the next charge taken in may explode because it does not mix with burned gas but with the weak charge in the clearance. The hot exhaust gases ignite the weak mixture which was rejected unburned to the exhaust at the previous cycle. If the ignition is imperfect, a good mixture may fail to explode and be exhausted, and may then be ignited in the exhaust pipe by the next exhaust of hot gases.

Adjustment of Point of Ignition. The proper timing of the ignition depends upon a variety of conditions of operation and construction of an engine. In the following paragraphs the various causes leading to the necessity for changing the relative point of

ignition are given, and the direction and amount of change necessary are indicated.

Adjustment for Change of Speed. In a high-tension system for a variable speed motor in which the alternations are produced by a trembler interrupter, the timer must be moved so as to interrupt the primary current earlier in the stroke when the speed of rotation is high than when it is low. This is due to the fact that there is a certain amount of lag in the ignition apparatus between the instant of first closing of the primary circuit by the trembler and the jumping of a spark between ignition points of the igniter. The time interval of this lag is constant whatever the speed of the motor, but the amount of movement of the crankshaft of the motor during the period of this lag is greater when the motor is running fast than when it is running slowly—at 1000 revolutions per minute, the crankshaft will move twice as far between the instant of first closing of the primary circuit and the jumping of the spark as it will when rotating at 500 revolutions per minute—and thus, to allow for this lag, the timer must be advanced as the engine speed increases, the amount of advance being proportional to the speed. In ignition systems whose interrupter is mechanically operated, as with magnetos, the lag is practically nil.

Adjustment for Rich and Lean Mixture. If a stationary engine operating on producer gas has its ignition properly timed for the mixture which has the highest rate of combustion while the producer is delivering rich gas, and the gas then becomes lean, the combustible mixture going to the engine will then become lean if the setting of the proportioning valves remains unchanged, and, therefore, the ignition will have to be advanced to secure the most efficient results for the kind of mixture then received. In the same manner, if the ignition is properly timed for the mixture which has the highest rate of combustion when the producer is delivering lean gas, the ignition will have to be advanced if the gas becomes very much richer and the setting of the proportioning valves is not changed.

Adjustment for Change of Compression. When an engine is throttle-governed the compression is varied by the governor action as the load varies. If such an engine is operating at full load with its ignition properly timed and the load falls off, the ignition must be advanced on account of the reduced compression and consequent

slower rate of combustion. When the load comes on again, the ignition must be retarded to its initial position.

Variation of Duration of Ignition with Change of Speed. Since the explosive mixture requires an appreciable amount of time for combustion, it must be ignited earlier in the stroke when the engine is running at high speed than at low speed, in order to obtain the maximum amount of power from the fuel consumed. The extent of the variation of the duration of ignition with variation of speed depends on the kind of ignition system employed and the location of the igniter, or igniters. A system which gives a spark of constant strength regardless of the speed of rotation of the engine must have the time of ignition varied more with variation of speed than is the case with a system which gives a stronger, or hotter, spark as the speed of the engine increases. If the igniter is located away from the body of the mixture in the combustion chamber—in the pocket over the valve, or between the valves of some types of engines—then the ignition must be more advanced than when the igniter is located near the center of the charge volume.

Most ignition systems operating on battery current give a spark whose strength is the same whatever the speed of the engine. On the other hand, most magneto-ignition systems using current direct from the magneto give a stronger spark as the speed of the motor, and consequently that of the magneto, increases. Less advance of the spark is therefore required in the magneto system in order to allow for the variation of intensity of the spark, an additional advance being necessary to allow for the lag if a trembler is used.

Less variation of the time of ignition is necessary if two or more igniters are used, located some distance apart in each combustion chamber. Aside from the greater certainty of ignition, the use of two igniters decreases the time required for combustion, since each of the propagating flames emanating from the points of ignition has less distance to travel than when ignition is at one point only.

Care and Adjustment of Ignition Systems. The ignition apparatus should always be kept clean and the insulation should be kept as free from oil as possible (especially in high-tension ignition), as oil destroys the insulating property of rubber.

Spark Plugs for Jump-Spark System. The spark plug for a

jump-spark system may be cleaned by the use of a stiff bristle brush and gasoline to remove the carbon deposits. The insulation of the plug should not be scraped with any hard tool, as the roughened surface resulting will afford easier lodgment for carbon deposits than a smooth surface. If a bead of metal has formed at the spark gap, or if the points and edges have become rough and irregular or burnt and pitted, they should be filed off square across the length of the wire, thus leaving sharp edges, from which a spark will jump more readily than from a smoothly rounded point. If, after filing the points, the spark gap is too great, or if the points are found in good condition but the gap is too great, the wire which forms one side of the gap should be bent enough to decrease the gap to the proper amount. In some plugs the insulated spindle or other part of the plug—depending on the design—must be adjusted to accomplish this. The width of the spark gap in a battery-ignition system should ordinarily be about $\frac{1}{32}$ of an inch, but a wider gap is sometimes used. Magneto makers generally recommend a spark-gap width, for magneto ignition, of from $\frac{1}{64}$ to $\frac{1}{50}$ of an inch for plugs and magnetos of the size ordinarily used on automobiles. In magneto-ignition systems for larger engines and those in which the spark plug and magneto are larger than on automobile engines, the spark gap is wider—about $\frac{1}{32}$ of an inch being suitable.

Igniter Plugs in Make-and-Break System. In make-and-break igniter plugs, the contact points, or contact surfaces if no contact pins are used, become pitted and uneven with use and should be dressed with a smooth-cut file so as to make good contact with each other. Care should also be taken to see that the points separate at least $\frac{1}{32}$ of an inch when tripped, otherwise the spark will continue to arc across the gap after the break, igniting the incoming charge during the next suction stroke, causing a back fire, as well as rapidly pitting and destroying the contact surfaces. A much wider separation than this is customary in make-and-break igniters, especially in large plugs. Sometimes persistent back fires can be traced to the fact that the current arcs across from the movable electrode to the cylinder wall after the igniter is tripped, due to the fact that the movable electrode comes too close to the wall. Lamps should be used in the circuit of each igniter to regulate the current flow and prevent the contact surfaces from pitting too rapidly from an excess

of current; one ampere has been found sufficient to give satisfactory ignition under all conditions.

Tremblers. Tremblers should be so adjusted as to use the minimum battery current that will give a satisfactory spark. Ordinarily, more current flows when the rate of vibration is high than when it is low The rate of vibration of the tremblers should be set somewhere near midway between the highest and lowest rates that will give ignition. If there is more than one trembler and coil, the tremblers should all be adjusted as nearly as possible to the same pitch. For the more usual sizes of coils, from one-half to one and one-half amperes is required for operating a four-cylinder high-speed motor.

Contact Timer. A sliding contact timer should have the rubbing surfaces well lubricated by means of either an oil bath or by packing the timer with soft grease. With a roller-contact timer it is generally better to use oil than grease, as grease is apt to prevent good electrical contact between the roller and the stationary contact piece; if grease is used, it should be very thin. A pressure-contact timer (in which the contact is made in the same way as in a trembler) is not intended to be submerged in oil and should not have any oil on the contact points, as it may prevent effective closing of the circuit when the contact points are covered with it. A soft bristle brush and kerosene should be used to clean the timer whenever dirt has collected in it to any appreciable extent and, after the cleaning, oil should be applied to the rubbing surfaces.

Setting the Timer. Remove the spark plug from the cylinder, leaving the wire connected, and ground the outer bushing. Rotate the crankshaft until the piston in the cylinder, from which the spark plug has been removed, is on dead center at the top of the compression stroke. Set the spark control at the position corresponding to ignition at dead center when the engine is rotating slowly—for high-speed motors not more than one-fifth of the entire movement of the spark control in advance of maximum retard. Uncouple the timer rotor shaft from its drive and rotate slowly in the direction in which it rotates while operating, until a spark jumps across the gap in the spark plug, which has been removed from the cylinder. Recouple the timer rotor shaft to its drive in this exact position. The general method here set forth applies equally well to make-and-break

ignition, only instead of removing the igniter plug the instant of firing can be determined from the snap of the tripping device.

Carbureter Adjustments. Close the gasoline needle valve and then open a few turns. Start the motor and allow it to run until it becomes heated to normal running temperature. If the carbureter has only one adjustment—the gasoline needle valve—open or close the needle valve until the motor fires regularly on all cylinders and runs at maximum speed for that throttle opening. A carbureter of this type is generally designed to give approximately equally good results throughout the range of speed and therefore the speed chosen for adjustment is not important. In a carbureter with an auxiliary air valve with high- and low-speed springs the upper or low-speed adjusting nut on the auxiliary air valve should be adjusted until the valve seats lightly before the motor is started, and at the same time the high-speed spring should have about $\frac{1}{16}$ of an inch play. After starting and heating up the motor the low-speed adjusting nut should be turned until the motor runs properly with the throttle closed. Then advance the spark and open the throttle and, if the motor back-fires, turn the lower or high-speed adjusting nut up or down until the back-firing ceases. The high-speed spring should always have at least $\frac{1}{32}$ of an inch play while the motor is running idle. This type of carbureter has no needle valve and a fixed gasoline nozzle, and if too much or too little gasoline is supplied the nozzle must be removed and one of a different size substituted.

If the auxiliary air valve has only one adjustment, the high speed, the low speed adjustment is obtained by adjusting the gasoline needle valve, and the high speed, by varying the tension on the auxiliary air-valve spring.

If the carbureter is fitted with an intermediate- and high-speed needle-valve adjustment, the method is to get the low-speed adjustment by means of the needle valve and the auxiliary air valve. Advance the spark and open the throttle until the needle-valve roller is on the track directly below the first or intermediate speed dial. Adjust the screw on this dial until the motor runs properly. Open the throttle wide and make the high-speed adjustment in the same way on the second dial.

When a second or auxiliary nozzle is used it comes into action only as the motor approaches high speed and should therefore be

so adjusted before starting that it cannot come into action at low speed. The low-speed adjustment is made as in one of the previous cases, depending upon the design. The high-speed adjustment is made by adjusting the opening of the auxiliary nozzle until the motor runs properly.

Fitting Piston Rings. Piston rings, when placed in the slot in the piston, should be of such a width that they rotate freely in the slot. In fitting snap piston rings, a clearance, a, Fig. 177, must be filed out of the slot, when both the cylinder walls and the ring are cold, sufficient in amount to allow for the expansion due to heat. This cold clearance must be such that when the engine is heated up to working temperature there will still be a slight clearance and the ends of the ring will not butt. If this happens, the wear on the cylinder walls or the rings, or both, will be very rapid. The following formulas are obtained from estimates, based upon practical experience, of the temperatures obtaining in the various parts of an engine.

The clearance to be filed in the slots of piston rings for *single-acting uncooled trunk pistons* may be taken as

$$a = 0.0060d \text{ to } 0.0075d \text{ in.} \qquad (6)$$

where d is the diameter of the cylinder in inches, a being the clearance space in the ring.

In the case of an uncooled trunk piston the first or innermost ring (nearest the explosion) is seriously heated, but, nevertheless, is not as hot as the uncooled piston body, since the ring is in contact with the water-jacketed cylinder wall. The temperature of the piston rings decreases very rapidly towards the outer end—at the last ring the difference between the mean temperatures of the ring and of the cylinder barrel will scarcely amount to more than 90° to 110° F. Nevertheless, all the rings of a set or of a size should be given the same clearance in order to make them interchangeable.

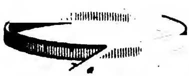
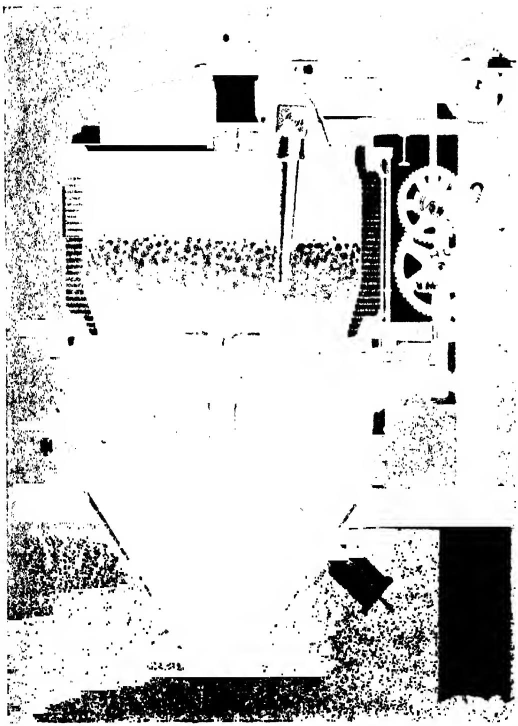


Fig. 177. Snap Piston Ring

For large double-acting water-cooled pistons the temperature difference supposedly approaches 0 degrees. Therefore a much smaller clearance a suffices—from 0.040 to 0.200 inches, depending upon the diameter of the cylinder.



SECTION THROUGH HUGHES GAS-PRODUCER—SELF-CLEANING TYPE WITH CENTER ASH DISCHARGE

Courtesy of Wellman-Seaver-Morgan Company, Cleveland, Ohio

PART II

GAS-PRODUCERS

INTRODUCTION

Producer Gas and Its Competitors. Producer gas is the gas resulting from the gasification of solid fuel where the heat required in the process is obtained by a partial combustion of the fuel itself; it is the most extensively used artificial fuel gas.

Natural gas is restricted to such a limited territory that its extensive use is out of the question. Retort gas requires a definite quality of coal, and a large, complicated plant, and makes a residue which must be disposed of. Coke-oven gas can be made only in a large, complicated plant, and requires the attention of a skilled chemist, and also a ready market for the coke. The water-gas process is intermittent, complicated, and not very efficient. The carbureted water-gas process, in addition to having the disadvantages of the straight water-gas process, requires oil for the carbureting and, moreover, the carbureting adds only illuminants which, in proportion to their density, add little to the heat value of the gas. Oil gas is restricted to a very limited territory, since it can be used commercially only where the cost of oil is very low. The use of blast-furnace gas is limited, as it can be obtained only at large iron works.

There has been a great demand for a gaseous fuel within the last few years, and this has given the producer-gas industry a sudden growth. This demand, and the resulting growth, are due not only to the advent of the gas engine, but also to the appreciation of the value of gaseous fuel for ceramic and metallurgical operations and the constant diminution of the natural-gas supply. High and easily controlled working temperatures, perfect combustion and high fuel economy are most readily obtainable by the use of a gaseous fuel—just the conditions required by many ceramic and metallurgical processes for successful operation. Many industries, which in the past have used natural gas for fuel, have started to use producer gas, as the cost of the natural gas has increased.

History of Producer Gas. About 1834, Faber du Faur, a German engineer, began using blast-furnace gas for heating furnaces. This gave such good results that the demand for the gas was greater than the supply furnished by the furnaces. From this he reasoned that it would be desirable to build a low type of blast furnace, omitting the charge of iron and using the furnace only for the production of gas to supply the increased demand. Circumstances prevented Faber du Faur carrying this idea into practice; but he announced it at that time, and several contemporary engineers began working on the problem.

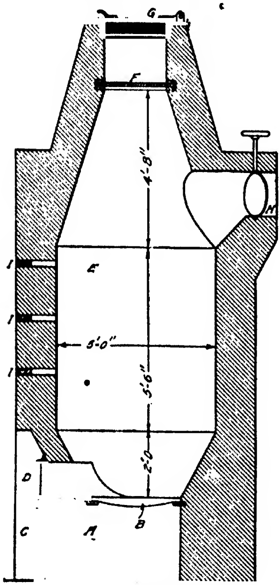


Fig. 1. Section of First Gas-Producer

First Gas-Producer. The first gas-producer was probably built by Bischof in 1839. It is shown in Fig. 1, and resembles a small blast furnace. *A* is the ash pit under grate *B*; *C* and *D* are cleaning doors, the former being made with openings to admit the air; *E* is the body of the producer; *F* and *G* are doors for charging the producer with fuel; *H* is the gas exit; *I* shows a peep-hole for examining the condition of the fire.

Later Developments. Ekman in Sweden, Wedding in Germany, Ebelmen in France, and Siemens in England, were also working on the problem between 1840 and 1860; and they all built certain types of producers. Ebelmen anticipated several present-day types of producers.

Siemens, Dowson, and Benier. The first producer to be used to any extent was the Siemens, which was introduced in England

about 1860. This forms the commercial starting point of the producer-gas industry. Dowson, in 1878, was the first to use producer gas for power purposes. The suction gas-producer was introduced in France, on a commercial scale by Benier, in 1895.

MANUFACTURE OF PRODUCER GAS

CHEMICAL AND MECHANICAL PROCESS

Firing to produce gas requires that the fuel bed be *thick* and *compact* enough to permit only a partial combustion of the fuel, so that a stream of combustible gas will be given off at the surface of the fuel. Direct firing requires that the fuel bed shall be sufficiently *thin* and *porous* to permit enough oxygen to get through the interstices in the fuel bed to produce vigorous combustion at the surface of the fuel.

Chemical Constituents of Producer Gas. *Simple Form.* The simplest form of producer gas consists of a mixture of nitrogen and carbon monoxide. That is, when a bed of charcoal or coke is blown with a dry air blast, the fuel bed will soon be at a white heat, when the following reaction will take place:

$$C+\underbrace{(O+N)}_{\text{air}}=CO+N \qquad (1)$$

In this formula the symbols represent the chemical elements entering into the reaction but do not show their relative amounts.

In case carbon dioxide is formed, it should be immediately converted into the monoxide, by an excess of incandescent carbon. Thus

$$CO_2+C=2\,CO \qquad (2)$$

The heat required for gasification is that which is evolved in burning the carbon to carbon monoxide. The heat available in the gas is that which will be evolved when the carbon monoxide is burned to carbon dioxide. The heat loss by this method is very high, as is shown by the following example:

1 lb. C burned to CO_2 evolves 14,500 B.t.u. = heat in fuel

1 lb. C burned to CO evolves 4,450 B.t.u. = heat lost

10,050 B.t.u. = available heat in gas (about 70 per cent).

Effect of Use of Steam. On account of the high heat loss, the use of simple producer gas is now obsolete. The use of steam not

TABLE I

Typical Producer-Gas Analysis

Constituents		Per Cent
Combustible	Hydrogen	8.0
	Marsh gas	3.0
	Olefiant gas	0.5
	Carbon monoxide	24.0
Condensible	Tar	1.0
	Water vapor	1.0
Diluents	Carbon dioxide	3.0
	Oxygen	0.5
	Nitrogen	59.0
		100.0

only reduces this loss, but also increases the heating value of the gas, and eliminates some of the difficulties of producer operation. Thus a small amount of water gas is made along with the producer gas. In some producers, the fuel undergoes a partial destructive distillation before going on to the fuel bed proper. Hence, modern producer gas is nearly always made in three processes, the best features of the retort, water-gas, and producer-gas processes being combined into one simple, continuous, and efficient process. This combination of the best elements of other systems is the secret of the extensive present-day use of producer gas.

Composition of a Representative Producer Gas. Producer gas is a mixture. Table I shows the composition of a representative sample of producer gas.

The proportion of each constituent present will depend upon the nature of the raw fuel, the type of producer, and the method of operation. Water vapor and tar, although generally present, are not usually determined and given in the analysis, since both will nearly always condense within a short distance from the producer. Tar in producer gas comes directly from the fuel; it will condense quite easily and will then be precipitated in the pipes.

A fixed or permanent gas is one which has no condensible constituents when the gas is cooled. Producer gas should be composed of fixed gases only, since the gas will always be cooled after leaving the producer and, if it contains any condensible constituents, these

will be deposited in the pipes; this will cause a heat loss and will also give trouble from the clogging of the pipes.

Typical Producer. A typical producer is shown in Fig. 2. It consists essentially of a steel jacket *A*; fire-brick lining *B*; support *C*; grate *E*; tuyère *F*, which is now frequently omitted in suction producers; air blast *H*; charging hopper *J*; poke-hole *K*; and retort or fuel reservoir *L*. In American types, *L* was usually omitted; but it

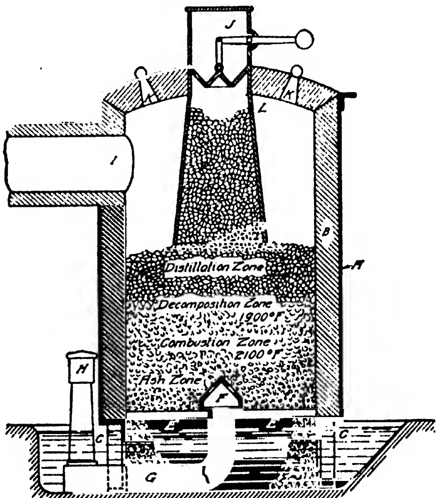


Fig. 2. Section of Typical Gas-Producer, Showing the Different Zones

is used quite extensively in European types and is now coming into use in the best types of American producers. Frequently, the grate is omitted and the fuel rests directly on the ash-pan bottom.

Steam Blowers. The steam and air should be introduced together so as to secure a thorough admixture. In a large number of producers, the air is forced into the producer by a steam blower, which is simply an air injector. Since a small quantity of steam must carry in a large quantity of air, the area of the surface of con-

tact between the two should be as large as possible, for the quantity of air delivered per minute by a steam jet depends upon the surface of contact between the air and the steam, irrespective of the steam pressure, up to the limit of exhaustion or compression that the steam jet is capable of producing. Two general types of steam blowers are shown in Figs. 3 and 4. The former has a very small area of surface contact between the air and steam and, as a result, this form of blower is very inefficient. The type shown in Fig. 4 will deliver several times as much air with a given quantity of steam as the blower shown in Fig. 3.

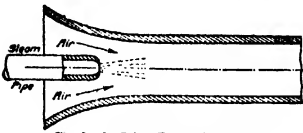


Fig. 3. Inefficient Type of Steam Blower

Chemical Action in a Gas-Producer. The chemical action in a gas-producer will be understood more readily by considering each successive step. Fig. 2 shows the fuel bed divided into four zones. In practice, the line of demarcation between the different zones is not always distinct, and they sometimes overlap one another.

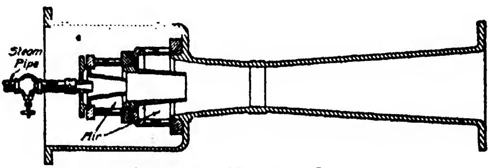


Fig. 4. Approved Type of Steam Blower

Ash Zone. No reactions take place in the ash zone, but it serves to protect the grate from the intense heat of the upper zones, and also pre-heats the air blast in an up-draft producer.

Combustion Zone. The combustion zone receives its name from the fact that the heat required for gasification is generated there by the combustion of the carbon, which burns to carbon dioxide. Thus

$$C+O_2=CO_2 \tag{3}$$

The intense heat generated there keeps the superimposed zone at its proper working temperature.

Decomposition Zone. The decomposition zone receives its name from the fact that the steam from the blast, and the carbon dioxide from the combustion zone, are there decomposed. Thus

$$H_2O \ + C = H_2 + CO \qquad (4)$$

$$(2+16)+12=2 \ +(12+16) \text{ molecular weights}$$

$$(1+\ 8)+\ 6=1 \ +(\ 6+\ 8) \text{ proportion of molecular weights}$$

$$CO_2+C=2CO \qquad (5)$$

The zone must contain an excess of incandescent carbon, and must be kept above 1800 degrees Fahrenheit, in order that these reactions may take place. Since the decomposition of the steam will absorb a large quantity of heat, it is evident that only a limited amount may be used, if the operation of the producer is to be continuous. By Equation (4) with 6 lb. of carbon burned to carbon monoxide, evolving $6 \times 4450 = 26{,}700$ B.t.u., 1 lb. of hydrogen will be separated from 8 lb. of oxygen, the two being in the form of 9 lb. of steam, and this will absorb exactly the same amount of heat that would be given off in the combustion of 1 lb. of hydrogen—namely, 62,100 B.t.u. The heat absorbed by the reaction will equal $62{,}100 - 26{,}700 = 35{,}400$ B.t.u. for every 9 lb. of steam decomposed, or 3933 B.t.u. for every pound of steam.

Distillation Zone. The distillation zone is so named because the heat from the lower zones effects a partial distillation of the fresh fuel in that zone. The addition of a charge of fresh fuel will always lower the temperature, and this will change the composition of the resulting gas. High temperatures in this zone are conducive to the formation of fixed gases, while low temperatures will be sure to produce a large yield of tar.

Working of Gas-Producer. The temperature of the gas as it leaves the producer should be kept low, or else the sensible heat loss due to the cooling of the gas between the producer and the place of use will be high.

The producer should be so arranged that the sensible heat of the gas may be utilized for pre-heating either the fuel or the air. The pipes for hot gas must be larger than those for cold gas, because of the larger volume per unit weight of gas at higher temperatures. Gas at 660 degrees F. has twice the volume of gas at 100 degrees F. The valves and dampers for handling hot gas must be water-cooled to prevent warping. Further, for gas-engine work

TABLE II

Effects of Temperature on Action of Steam

Temperature F.	Percentage of Steam Decomposed	Gas Analysis		
		CO_2	CO	H
1245	8.8	29.8	4.9	65.2
1750	70.2	6.8	39.3	53.3
2057	99.4	.6	48.5	50.9

This table shows the importance of keeping the temperature of the combustion and decomposition zones near 2000 degrees Fahrenheit, if satisfactory results are to be obtained.

the sensible heat is of no value, and the gas should be cooled when it goes into the engine cylinder, in order to increase the charge weight.

Pre-heated air will not only reduce the heat losses, but will also induce better gasifying conditions. The waste heat in the gas-engine exhaust may be used for pre-heating the air when a producer furnishes the gas for the engine. By such an arrangement the efficiency of the producer can be increased ten per cent.

Satisfactory operation of the producer can be secured only when the different zones are kept at their proper temperatures. Temperatures which are too low result in the formation of small amounts of carbon monoxide, and large amounts of carbon dioxide and water vapor, causing a heavy heat loss, since the last two are not only diluents, but also represent a certain heat loss in the producer. An excess of steam causes a reduction of temperature. The effect of different zone temperatures on the composition of the gas is shown in Table II, the data for which has been taken from an actual test.

The decomposed steam acts as a carrier of heat energy between the producer and the chamber in which the gas is to be burned. All the heat absorbed from the producer in the decomposition of the steam and in the formation of hydrogen will be given out when the hydrogen is burned back to water. That is, when steam is used, a certain amount of sensible heat that would otherwise be wasted in the producer-gas process is locked up temporarily in the form of hydrogen, and carried over into the combustion chamber, where it becomes available. Under no circumstance can the use of steam cause a

gain of heat, and the tendency will always be to lower the temperature of the fire. In addition to the conservation of the heat losses in the process of gasification, the steam has a very desirable mechanical effect on the fuel bed, by softening the clinkers, preventing localized combustion and the fusing of the clinkers to the brickwork, keeping the fuel bed porous and homogeneous, and protecting the grate by keeping the intense combustion away from it.

The producer should be supplied with all the steam that it can decompose, in order to secure a high efficiency. This maximum quantity will vary with the nature of the raw fuel, with the type of producer, and with the method of operation.

Fuel. Practically every known solid fuel has been successfully used for the manufacture of producer gas. The purpose for which the gas is to be used, and the type of producer, will, however, determine what fuels may be used in each particular case. Since each fuel will give the gas certain definite properties, it is evident that the producer gas made from different fuels may vary perceptibly in composition. Producer gas made from bituminous coal will be high in easily condensible hydrocarbons, generally spoken of as "tar"; while that made from anthracite coal will have a low percentage of tar. Thus, some fuel with a certain type of producer might make a quality of gas that would give good results in a steel furnace; while this same gas might be worthless for use in a gas engine. Impurities in the raw fuel will, in certain cases, give the resulting gas certain constituents that would make it unfit for certain kinds of work. In burning certain ceramic products with producer gas made from fuel containing volatile sulphur compounds, ammonium salts, or other impurities, considerable difficulty may be experienced from the action that the gas may have on the particular product under treatment. On the other hand, if a muffle kiln—that is, one in which the combustion products do not come in contact with the ware that is being burned—is used, the impurities would not make any difference.

The size and condition of the fuel are of considerable importance. A crushed coal will always give better results than coal with large lumps. Some run-of-mine coal is now being used in gas-producers; but it must be remembered that not all things that are possible are desirable. The use of fine dust, also, is not good prac-

tice. The fuel should be dry, since any moisture that it contains must be driven off in the producer, and this will cause a certain heat loss. Anthracite, bituminous, and brown coal, peat, lignite, wood, sawdust, shavings, tanbark, and similar refuse have all been used for making producer gas.

GASIFICATION LOSSES

Efficiency. Since producer gas is made by a partial combustion of the fuel itself, it is evident that there must always be a certain loss in the process of gasification; and, as a result, the efficiency of the gas-producer will always be less than unity.

$$\text{Efficiency}=\frac{\text{Heat units in gas from a unit weight of fuel}}{\text{Heat units in a unit weight of fuel}}$$

With a properly designed and carefully operated gas-producer, the gasification loss should not be over 20 per cent; that is, the efficiency should be at least 80 per cent. Thus, if the fuel contained 14,000 B.t.u. per pound, and the gas evolved from a pound of that fuel contained 11,200 B.t.u., then the efficiency would be equal to $\frac{11,200}{14,000}=.8$, or 80 per cent.

The heat energy in the gas will be of two forms—heat of combustion and sensible heat, by virtue of the temperature of the gas. Since the latter will be lost if the gas is cooled down to atmospheric temperature, it is evident that a gas-producer will have two efficiencies, depending upon whether the gas is used hot or cold. The former is called the *hot-gas efficiency*, while the latter is called the *cold-gas efficiency*.

If any carbon passes through the producer without being converted into gas, a correction must be made to get the true efficiency of the gasification. The grate efficiency represents the percentage of carbon actually gasified. A grate efficiency of 96 per cent means that 96 per cent of the carbon charged into the producer is gasified, and 4 per cent passes out with the ashes. The true efficiency of the producer or the efficiency of gasification of the fuel will be the efficiency as first determined divided by the grate efficiency. If the efficiency as first determined—or, what is the same thing, the apparent efficiency—is 80 per cent, and the grate efficiency is 96 per cent, then the true efficiency of the producer is 80 per cent ÷ 96 per

cent or 83.3 per cent. This refinement is seldom applied, since the producer and grate should be so designed that no unburnt carbon is carried out in the ash.

Heat Losses. There are many heat losses in the process of gasification. By judicious management, each of these may be reduced to a very small quantity. The principal losses are the sensible heat loss and the carbon dioxide loss.

The *sensible heat loss* is the heat carried out by the gas, by virtue of its temperature. If the temperature of the exit gases is 1000 degrees Fahrenheit, this loss will amount to about 11 per cent. If the producer gas is high in hydrogen—which has a high specific heat—the percentage of loss will be higher. The sensible heat loss is large in nearly all forms of producers. The loss due to *carbon dioxide* is frequently high; in bad cases it may amount to 10 per cent.

Heat Balance. The principal items in the heat balance of a gas-producer are as follows:

Debit Side		Credit Side
Calorific power of fuel		Calorific power of gas
Heat in air blast		Lost in ashes
Heat in steam blast		Lost in unburned carbon
		Lost in tar and soot
		Lost in sensible heat of gas
		Lost in heating undecomposed steam
		Lost in evaporating moisture in fuel
		Lost in volatilization of hydrocarbons
		Lost in radiation
Sum of debits	=	Sum of credits

REPRESENTATIVE TYPES OF GAS-PRODUCERS

The prefixing of either the name or the type of the producer to the gas made therein is not to be recommended, that is, names such as "Siemens gas"; "Dowson gas", "Mond gas", and "suction gas" are undesirable. It was originally thought that each design or type of producer would make a gas with a certain distinctive quality. It is true that the gas made in different producers will vary in composition; but this variation is due to the method of operation or to the nature of the raw fuel used, rather than to the type of producer.

Gas-producers may be defined as follows:

A *water-seal gas-producer* is one so constructed as to have a seal of water between the interior of the producer and the air.

A *continuous gas-producer* is one that may be operated continuously for a long period of time. To secure this condition, the fuel must be introduced, and the ashes removed, in such a manner as not to interfere with the process of gasification.

An *up-draft gas-producer* is one that introduces the air at the bottom and removes the gas at the top.

A *down-draft gas-producer* is one that removes the gas from the bottom, and introduces the air blast at the top of the fuel bed, and in this way causes the draft and the resulting combustion to go downward. The term *inverted-combustion* is also used synonymously for *down-draft.*

A *by-product gas-producer* is one that, in addition to the regular production of gas, makes one or more auxiliary products based on certain constituents of the raw fuel or resulting gas, constituents that, generally, would otherwise be lost.

An *underfeed gas-producer* is one in which the fresh fuel is fed into the bottom of the producer.

Classification. All gas-producers may be classified into the following groups:

(1) Suction
  (a) Draft induced by a chimney
  (b) Draft induced by an exhauster
  (c) Draft induced by a gas-engine piston
(2) Pressure
(3) Balanced draft

Suction Producer. A suction producer is one in which the resulting gases are drawn or induced away from the producer; the interior of the producer is kept at less than atmospheric pressure, and the air is forced in by the pressure of the outside air. The suction may be obtained by a chimney, exhaust fan, or gas-engine piston. There has been considerable confusion in nomenclature, some proposing that the only apparatus that should be called a suction producer is that in which the suction is furnished by the engine piston. This would be quite correct if it were applied to the *plant*, i.e., if such a plant were called a *suction gas power plant.* To the producer itself, it is immaterial how the suction is produced. An engine of any size can be made to furnish the suction for the producers and the plant will run satisfactorily. It has been found

better, however, from the standpoint of economy, and to obtain greater reliability, to substitute some other apparatus—an exhauster—for the gas-engine piston in large gas-engine plants, since an exhauster is a much more efficient gas pump than is the engine piston. The only advantage in having the suction furnished by the engine piston is its simplicity; this is important only to small plants where simplicity throughout is a prime requisite. In large plants, however, the greater economy and reliability which result from the use of an exhauster always more than overbalance the loss of simplicity.

Pressure Producer. A pressure producer is one in which a mixture of air and steam is supplied to the producer under pressure, the pressure being sufficient to force the mixture through the fuel bed and to force the resulting gases to the point where they are to be used.

Balanced-Draft Producer. A balanced-draft producer is one in which a combination of the two previous methods is used. A mixture of air and steam is furnished to the producer, under pressure, by a blower, and the resulting gases are drawn off by an exhauster and sent to the engine under pressure—the relative pressure and suction being so adjusted that the pressure on top of the fire is just atmospheric. The advantage of this system is that the fire can be properly looked after and cleaned without the leakage either of gas into the producer room or air into the producer. This is particularly advantageous for continuous operation, since the fire needs occasional attention and poking to clean out the ash and to prevent the formation of ash chimneys, clinker, or of fire arch. In a pressure producer, when a poke-hole is opened, gas flows out into the room and the operator will not give the fire proper attention because of the discomfort from the gas. In a suction producer, when a poke-hole is opened, air flows into the producer and as soon as it comes into contact with the hot gas, the gas catches fire and burns, thus increasing the amount of inert gases and weakening the gas delivered to the engine; if the poke-hole is kept open too long this will become so serious as to give trouble in operating the engine. In a balanced-draft producer, on the other hand, since there is no difference of pressure between the inside of the producer, on top of the fire, and the producer room, there will be no flow in either direction.

SUCTION TYPE

Small Producers in Which Engine Furnishes Suction. *Fairbanks-Morse Type.* Small producers of this type are used entirely for power purposes—furnishing gas for gas engines. The producer shown in Fig. 5 is typical of this class. The producer, or generator

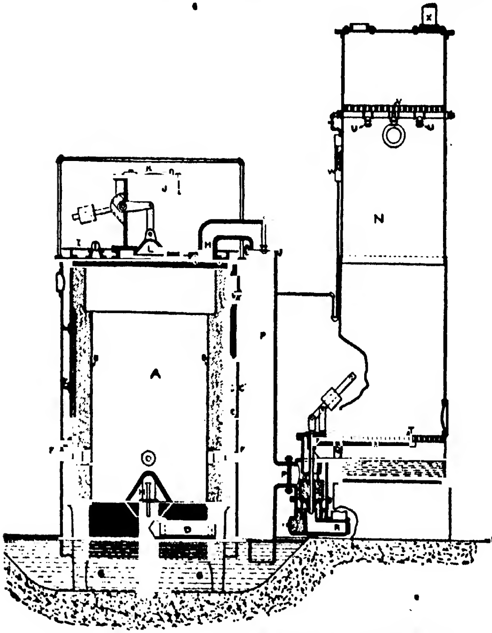


Fig. 5. Cross Section of Fairbanks-Morse Suction Gas-Producer Arranged to Show Principal Parts
Courtesy of Fairbanks, Morse, and Company, Chicago

proper, is constructed with a double-walled shell *C* and *C′*, between which the current of inlet air flows, picks up the heat radiated through the walls, from the producer, and returns it to the fuel bed through the pipe *D*. The vaporizer is located within the double

walls and is in the form of troughs E, into which a measured stream of water runs, depending upon the suction in the producer induced by the engine, i.e., upon the load. The water is vaporized by the radiated heat, mixes with the heated air and passes through the fuel bed, breaking up the clinker and cooling the fire to working temperature. The producer has a water-sealed bottom, the water being carried in the ash pit G. The fuel bed is not carried upon grates, being supported upon a bed of ashes in the ash pit, which, as they accumulate above the sight holes F can be removed through the water-sealed ash pit. The air and steam blast is admitted to the center of the fuel column through the bonneted pipe, or blast hood H. The shell C'' is well lagged and the top I of the generator is water-cooled to prevent radiation to the producer room and to keep the top of the producer cool enough so that the operator can properly attend to it without discomfort. The inner shell C' is protected by a fire-brick lining B. The charging hopper J is sealed top and bottom by the cover K and counterweighted valve L, respectively, and is so constructed that it is impossible to open either one if the other is not closed. The bottom valve L serves also to distribute the fuel over the surface of the fuel bed in charging. Gas is drawn off from the top of the producer by the pipe M. In the top of the downcomer P, leading to the scrubber N, is located a spray nozzle through which flows a very small quantity of water. This water is vaporized by the hot gas and aids in the cleaning when it is again condensed in the scrubber, as the particles of water in condensing steam are smaller than can be mechanically obtained and are thus able to envelop and weigh down the smaller particles of impurities in the gas, which would otherwise not be removed from the gas. The relatively large mechanically obtained particles of water are too large to envelop the small particles of dust in consequence of the increase of skin tension with increase of size. The producer and scrubber or producer and purge pipe R are connected by the water-sealed three-way valve O in such a way that the scrubber cannot be in communication with the purge pipe.

Syracuse Suction Type. The producer shown in Fig. 6 is also a double-shell producer in which the air is brought down to the fuel bed through the space between the shells and is there pre-heated. The inner lining is protected by fire brick only to the height at

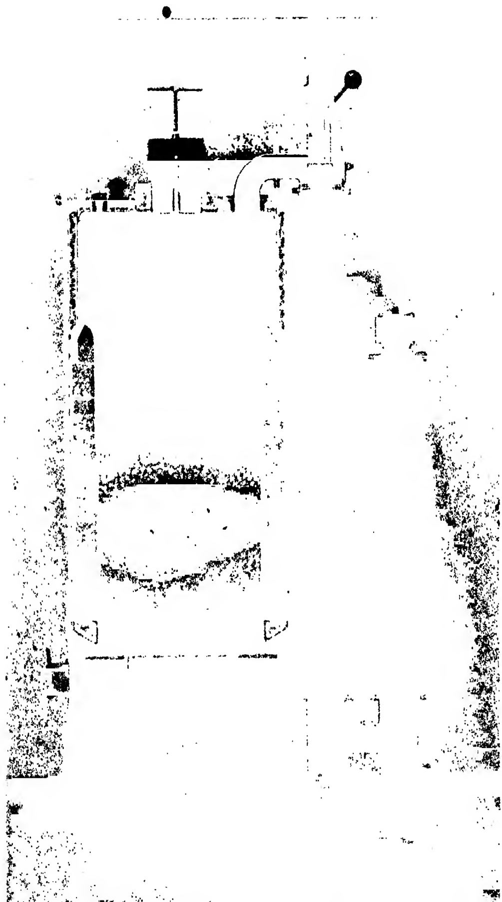


Fig. 6. Syracuse Suction Gas-Producer Shown in Section
Courtesy of Syracuse Industrial Gas Company, Syracuse, New York

which the fuel bed is ordinarily carried. Above this level, the sides and top are protected by a water jacket which also serves as the vaporizer. The top of the ash pit is cast to form an annular trough which catches any overflow from the wall jacket and serves as a supplementary vaporizer. This vaporizer also serves to improve the quality of the gas when starting the producer. Ordinarily,

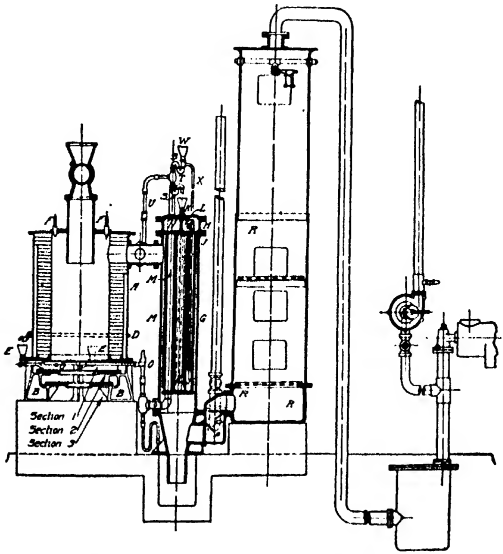


Fig. 7. Section of Crossley New Type Patent Suction Gas Plant
Courtesy of Crossley Gas Engine Company, Manchester, England

when a producer is started up, it is blown until the quality of gas generated is good, and the engine is then started. The rate of gasification in blowing up is much lower than in normal operation. When the engine is started, the pull on the fire is much increased, the conditions in the producer have not had time to become stable, and the quality of the gas drops for a short time and then gradually

builds up as the producer approaches normal operating condition. After a stand-by the vaporizer at the top is cool while the ash pit is hot, so that with this supplementary vaporizer the entering air will carry sufficient steam, upon starting up, until the top vaporizer is heated up. No blast pipe and hood are used in this case, the distribution being obtained by carrying the fuel bed on an inverted-cone or a flat grate. The grate-bar design and spacing depend upon the character of the fuel to be gasified. The charging hopper is

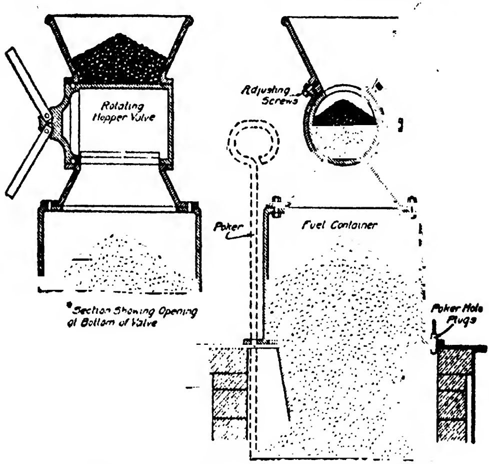


Fig. 8. Rotating Charging Valve of Crossley Suction Gas-Producer

sealed at the bottom by a rotating metal slide and at the top by a sheet-metal cover, the lower edges of which are water-sealed. This cover is so made that it cannot be removed until the lower valve is absolutely closed. The purge valve is water-sealed and at the same time water-cooled, and is located at the top of the scrubber, which takes the place of the downcomer in the previous producer.

Crossley Suction Type. The English gas-producer shown in Fig. 7 has a vaporizer entirely separate from the generator, which permits the use of an open-hearth type of stepped grate and conse-

quently permits the condition of the fire to be examined without opening any fire door or in any way altering the conditions of operation. The plates composing the stepped grate are so disposed as to be outside the angle of repose of the fuel, which insures that none of the fuel will fall from the grate. Below the bottom plate the fuel rests on its own bed of ashes. Clinker, which has been poked down through the holes *F F* in the top plate, can be removed at the step grates. A small stream of water delivered onto the top grate plate overflows onto the lower grate plates and keeps them cool at the same time that it supplies steam for gas making. The rotating charging valve is shown in Fig. 8; it insures that no air gets into the generator as a result of feeding fresh coal. In the vaporizer shown in Fig. 9, the water passes in succession down the inner and up the ribbed outer tubes from right to left

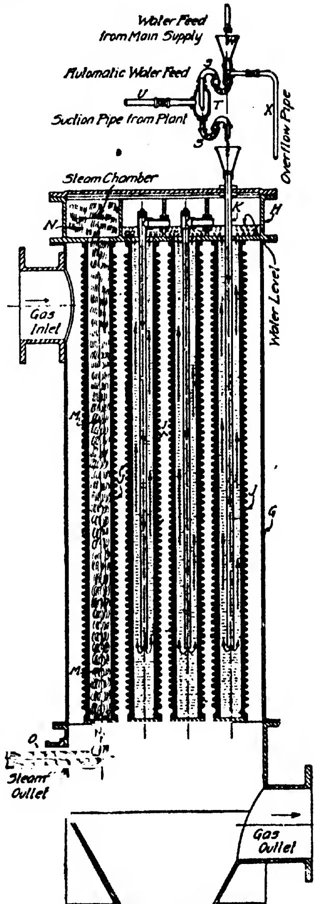


Fig. 9. Section of Crossley Vaporizer and Automatic Water Supply Device

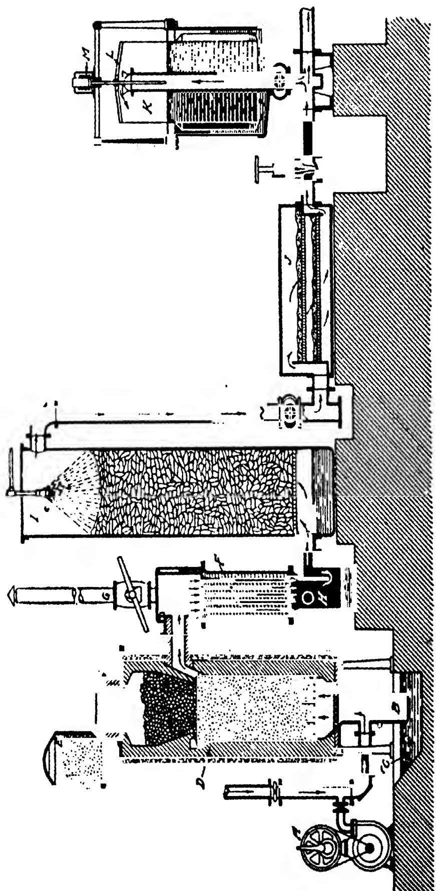


Fig. 10. Pintsch Suction Gas-Producer—Typical for 50-Horsepower Plants and Larger

and goes out, finally, as superheated steam. The water supply is regulated by the automatic device shown, which is operated by the suction at the generator; no water can flow when the engine is not working and the amount of water going to the vaporizer increases with the suction.

Pintsch Suction Type. The producers shown in Figs. 10 and 11 are of a different type of construction. Referring to Fig. 10, for larger-sized installations, *A* is the hand-blower. *B* is the ash

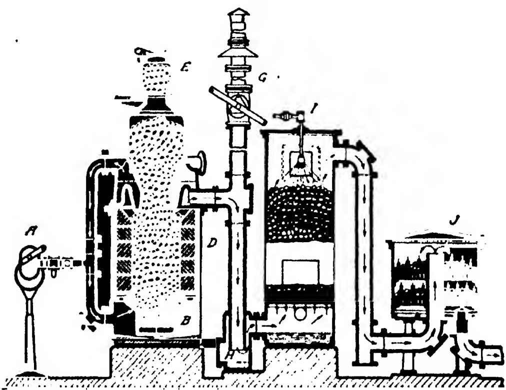


Fig. 11. Pintsch Suction Gas-Producer for 40-Horsepower Plants and Smaller
Courtesy of Seager Engine Works, Lansing, Michigan

chamber with water seal *C*. *D* is the body of the producer with charging hopper *E*. *F* is a tubular vaporizer above which is the vent pipe *G*. *H* is a settling chamber. The air for the producer is drawn through *F*, in that way absorbing the steam formed in the vaporizer, and is then taken to the ash chamber by means of a pipe not shown in the illustration. *I* is an ordinary tower scrubber filled with coke and supplied at the top with a spray of water. *J* is a purifier; the two shelves are filled with shavings, sawdust, or some similar material; as the gas passes through this, some of the impurities are filtered out. *K* is an automatic regulator; it consists essentially of a tank of water containing the bell *L*, which is supported by the spring *M*. The object of the device is to make the actual time

of drawing gas away from the producer of longer duration than the time occupied by the gas-engine piston in charging the engine cylin-

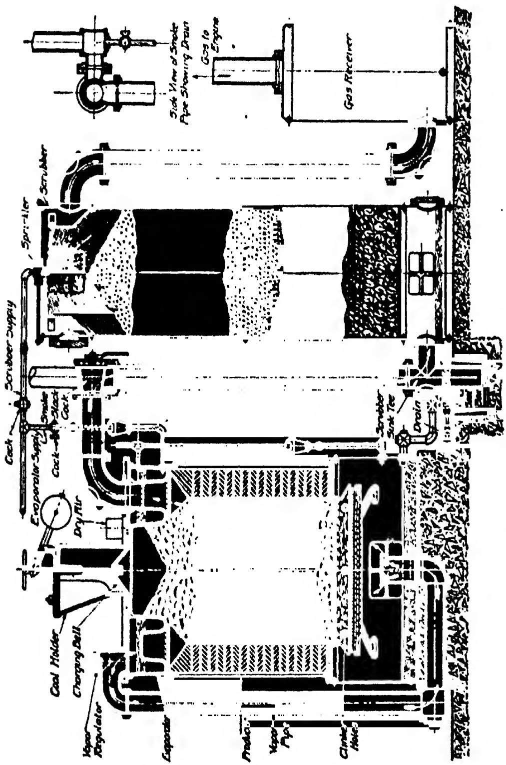


Fig. 12. Cross Section of Otto Suction Gas-Producer
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

der with gas. The operation is as follows: When the gas-engine piston draws gas to fill the cylinder, about half will be drawn from the chamber *K*; as a result, the exterior atmospheric pressure will

cause L to move down and compress the spring M. Just as soon as the engine stops drawing gas, the spring M will draw L back to its original position, and the gas required to fill K will be drawn from the producer. In this way the process of gasification is carried on after the engine piston has filled the engine cylinder.

The producer shown in Fig. 11, for smaller powers, is similar to the above with the exception that the vaporizer is an internal-ring vaporizer instead of an external apparatus.

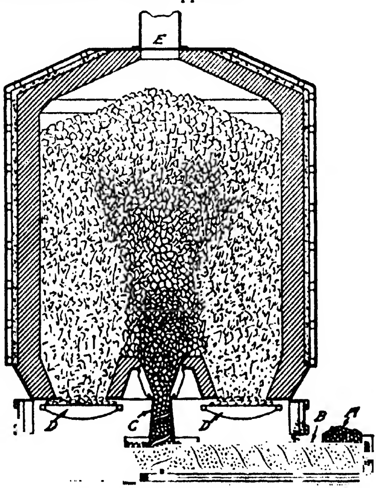


Fig. 13. Capitaine Underfeed Gas-Producer

Otto Suction Type. The cross-sectional view, Fig. 12, of a small suction producer with a fixed grate is self-explanatory.

Large Producers with Suction Furnished by Power-Driven Exhauster. If bituminous coal is used in a gas-producer, heavy condensible vapors are distilled off. These, unless they are decomposed and converted thereby into permanent gases, will condense in the form of tar as soon as they arrive at a cool place. The hydrocarbons have great heating value; and if they are not utilized in the producer, the efficiency of the plant is lowered. They are, however,

a source of considerable trouble and annoyance. The tar must be separated from the scrubber water before this water can be permitted to go into a sewer; and if the condensation and separation of the tar are not complete, it will cause trouble in the engine by depositing on the valves.

Many gas-producers are designed for dealing with bituminous coal in such way as to decompose and partly burn the hydrocarbon vapors and convert them into permanent gases. This can be accomplished either by an *underfeed* or by a *down-draft* producer.

Capitaine Underfeed: The Capitaine underfeed suction gas-producer is shown in Fig. 13. The coal is introduced at *A* and is then fed over to the center of the producer by spiral conveyor *B*, which delivers the coal to the vertical spiral conveyor *C;* this, in turn, screws the coal up into the center of the fuel bed. The ashes are worked out through grates *D*, while the gas is withdrawn at *E*. The primary object of this design of gas-producer is to introduce the fuel in such a manner as to secure a slow agitation of the fuel bed and also compel the volatile products of the green fuel to pass up and through the mass of superimposed incandescent fuel; in this way the volatile matter will be converted into fixed gases. By comparison with Fig. 2 it will be seen that the distillation zone is under the fuel bed in Fig. 13.

Loomis-Pettibone Bituminous Type. In the producer or generator shown in Fig. 14 the draft travels in the reverse direction from that in any of the producers so far shown, i.e., a down-draft producer. The fuel bed in the producer *A* is supported on a series of interlocking fire-brick arches, which serve the purpose of a grate. Above and below this grate are located cleaning and ash-pit doors, which are made of cast iron lined with fire brick. At the top of the producer, at the center of the cross section, is located a cast-steel water-cooled air-inlet nozzle provided with a cast-iron connection to the top of the economizer *B*, and provided, also, with a swinging door to enable inspection of the fire. The annular space between the air-inlet nozzle and the fire-brick lining forms a fuel reservoir, which, since the draft is down, is simply provided with single doors for the introduction of fresh fuel. Any leak of air into the producer while the charging doors are open will simply take the place of air which otherwise would come from the nozzle, and the quality of the gas

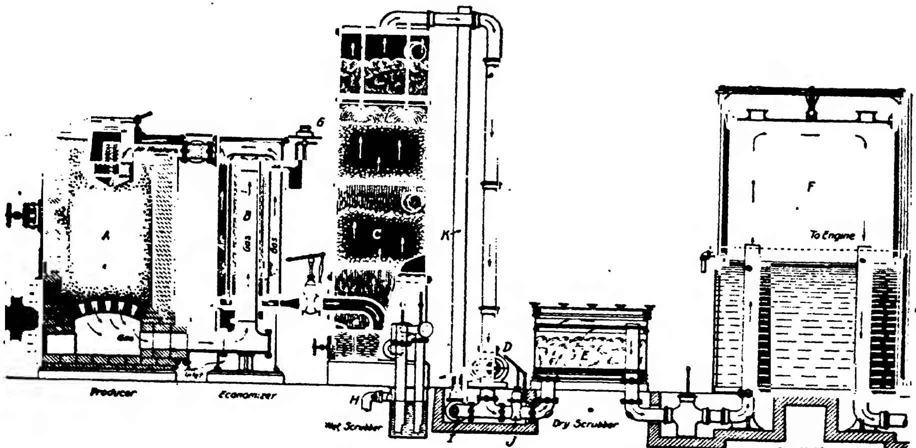


Fig. 14. Loomis-Pettibone Suction Gas-Producer for Bituminous Coal

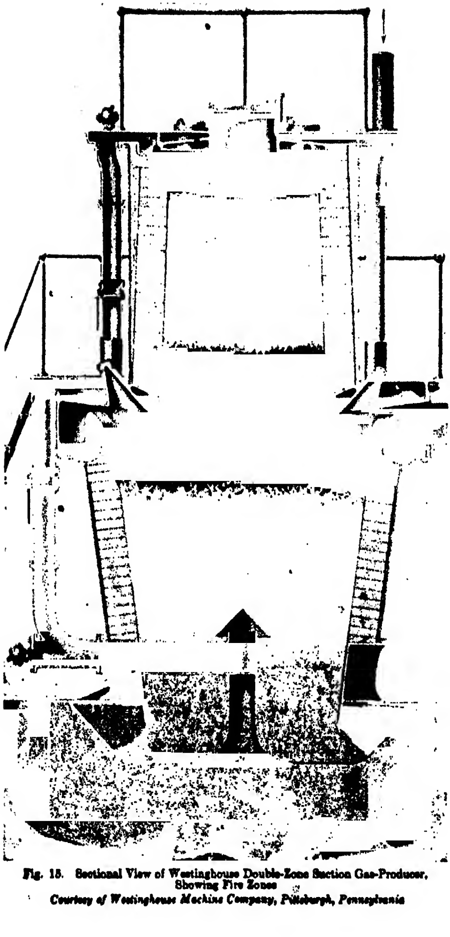


Fig. 15. Sectional View of Westinghouse Double-Zone Suction Gas-Producer, Showing Fire Zones

Courtesy of Westinghouse Machine Company, Pittsburgh, Pennsylvania

will be affected only from the fact that no steam is admitted in the air that leaks through the open doors. The gases distill gradually out of the coal as it descends and is slowly heated. These gases pass through the whole depth of the fire, and are thereby heated to such temperature as partly to burn, and partly to decompose, the tar vapors. A fire-brick-lined connection at the base of the producer leads to the economizer, or combined vaporizer and air pre-heater. The economizer, shown at B, is of the vertical return-tubular type, in which the hot gases pass from the base upward through a large central wrought-iron tube, the upper end of which is attached to a flanged diaphragm. Near the outer edge of this diaphragm is fitted a nest of return flues of relatively smaller diameter, to conduct the hot gases downward to the outlet casting at the base leading to the wet scrubber. The central wrought-iron tube in the economizer is fitted near its upper end with a small basin from which the water, fed in automatically proportioned quantities by the vacuum bell mechanism or water regulator shown at G, flows down over the central tube, the function of which is that of a flash boiler. The cool air, entering the base of the economizer shell and passing up around the tubes, carries along with it this evaporated water and enters the top of the producer in a highly heated condition. The suction in the scrubber and vaporizer is produced by the cycloidal impeller-type exhauster D, which forces the gas from there into the gas holder F, from whence it flows to the engine as it is needed.

Westinghouse Double-Zone Suction Type. Another method of converting the hydrocarbons is to gasify bituminous coal in a double-zone producer, an example of which is shown in Fig. 15. This producer is in reality composed of two producers—a down-draft producer mounted on top of an up-draft producer, with the gas taken off at the junction of the two. Coal is charged at the top through a suitable opening in the water-cooled cover plate. Through this same plate air is admitted and drawn through the fuel by the suction at the gas offtake. The amount of air supplied is insufficient to support complete combustion, but enough to cause coking, and the heat of this process distills the hydrocarbons from the green coal. As these pass through the coking zone, they are broken down into the more stable methane, or marsh gas, and carbon. Before the fuel reaches the level of the offtake, the available oxygen has been

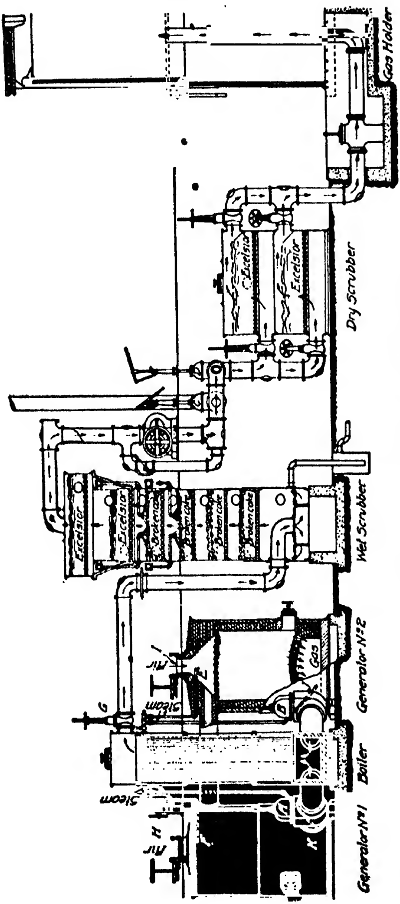


Fig. 16. Cross Section of Loomis-Pettibone Gas Generating System
Courtesy of Power and Mining Machinery Company, Milwaukee, Wisconsin

consumed so that complete combustion is impossible. In the middle of the producer, there is, therefore, a body of unburned coke. As this passes farther downward, it reaches a zone to which air is supplied from below through a tuyère and the coke is then gasified as in any up-draft producer. As the air passes upward through the lower part of this zone, complete combustion takes place, resulting in carbon dioxide CO_2, but as this gas rises, it unites with the carbon of the hot coke and forms carbon monoxide, CO. As the carbon monoxide is formed, it passes to the offtake where it mingles with the gases descending from the upper fire zone. The ash resulting from the combustion of the coke passes through the open bottom of the producer shell into the water-sealed ash pit.

Large suction producers in which the suction is furnished by an exhauster are also built for the gasification of anthracite.

Loomis-Pettibone Intermittent Type. The producer shown in Fig. 16 is designed to handle bituminous coal and is of the intermittent type. In normal operation the draft is downward and producer gas is made. Because of the character of the fuel and the incandescent zone necessary to fix the hydrocarbons, clinker is very readily formed during this period until it becomes so bad as to interfere with the draft. The air is then shut off from the producers and they are blown with a steam jet, no air being admitted. This produces pure water gas and is continued until the fire is chilled off and the clinker broken up into small particles and thus the resistance to the draft is reduced. With a fresh fire the producer-gas runs can be of long duration and the water-gas runs of very short duration, but the longer the producer is run without cleaning out entirely, the shorter are the producer-gas runs and the longer the water-gas runs. The condition of the fire may get so bad as to allow producer-gas runs of only ten or fifteen minutes and requiring water-gas runs of as much as a minute or two, whereas, with a clean fire, the producer-gas runs can be made as long as several hours and the water-gas runs seldom exceed thirty seconds In normal operation (producer-gas run) the exhauster creates a downward draft in both generator *1*, shown in elevation, and in *2*, shown in section, with the top doors *H* and *I*, and valves *A*, *B*, *G*, and *D* open. When the producers are being blown up to start, valve *D*, leading to the holder, is closed and valve *C*, to the purge pipe, is opened. As

soon as the fires are thoroughly kindled, and during all producer-gas runs, steam is admitted to the tops of the generators by means of the pipes *F* and *E*, and is mixed with the air drawn through the open top doors. The resulting producer gas is drawn down through the grates and ash pits of generators *1* and *2*, and passes up through the vertical boiler, which acts as an economizer by abstracting some of the sensible heat in the gas and generating steam with it to be returned to the producer. The steam is mixed with air during the producer-gas run, and is used for blasting the fires during the water-gas run. After leaving the boiler the gas is drawn under suction through the wet scrubber, or tower washer, passes through the exhauster, is forced through the dry scrubber under pressure, and from there to the gas holder, from whence it is drawn as needed. When a water-gas run is to be made, the top doors *H* and *I* and valve *B* are closed, and steam is blown up through the fire in generator *2* by admitting it to the ash pit by means of the pipe *J*. The resulting water gas and steam is blown to the top of generator *2*, from there across to the top of *1* by the brick-lined connecting gas pipe, down through the fire in generator *1*, and out through valve *A* into the vertical boiler as in normal operation. In alternate water-gas runs the valve *A* is closed, valve *B* remains open, and steam is introduced into the ash pit of generator *1* by means of pipe *K*. The gas holder is of sufficient capacity to mix the producer and water gas, so that the resulting mixed gas is never of a higher heat value than can be handled by the engine, and to insure that a charge of pure water gas cannot be drawn by the engine.

The principal difficulty with this type of producer is the fact that it is not capable of continuous operation, as it must be shut down and the ash and fuel bed entirely drawn at intervals of about six days. Because of the high heat in the fuel bed and the high ash content of the fuel ordinarily used, this is a very arduous task. This type of producer is used for both power and fuel-gas purposes, firing furnaces or kilns.

PRESSURE TYPE

Syracuse Bituminous Pressure Type. The producer shown in Fig. 17 is designed to handle bituminous coal for fuel-gas purposes. The charging hopper is so arranged that the gas given off the fresh fuel when fired, and which contains most of the hydrocarbons,

does not pass directly to the burners but is re-circulated and enters the fire with the main supply of air and steam. As the gas strikes the incandescent fuel, the hydrocarbons are changed to a permanent gas which will not condense in and clog the flues. The amount of

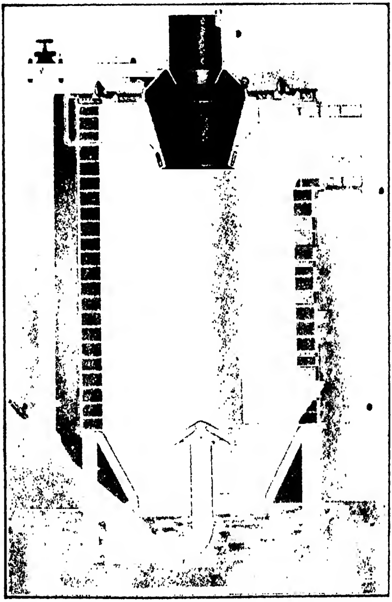


Fig. 17 Syracuse Bituminous Pressure Gas-Producer
Courtesy of Syracuse Industrial Gas Company, Syracuse, New York

gas so re-circulated is regulated by means of the valve shown at the top of the producer.

An additional advantage is that the top of the fuel bed remains even and the heating value of the gas does not vary when coal is fired, as occurs when the ordinary producer is used. A rotary grate is also sometimes used in this producer.

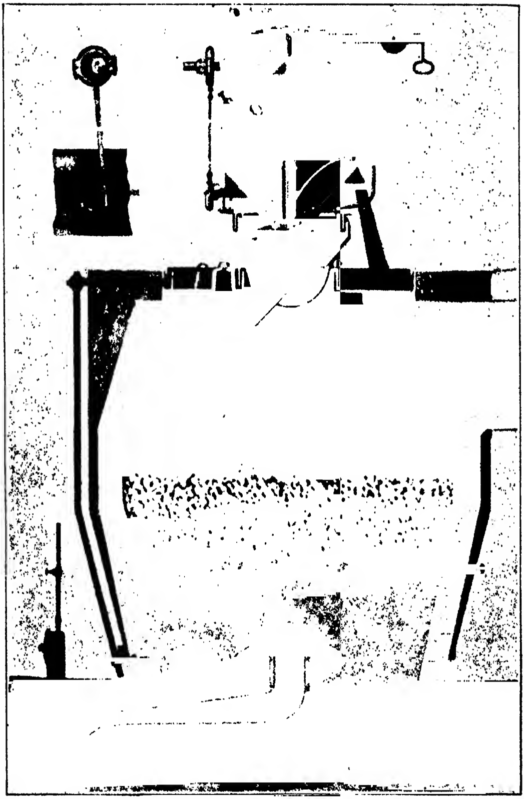


Fig. 18. Section of Morgan Continuous Gas-Producer with George Automatic Feed
Courtesy of Morgan Construction Company, Worcester, Massachusetts

Morgan Continuous Type. A producer which is largely used for furnishing gas to continuous-heating furnaces for supplying hot

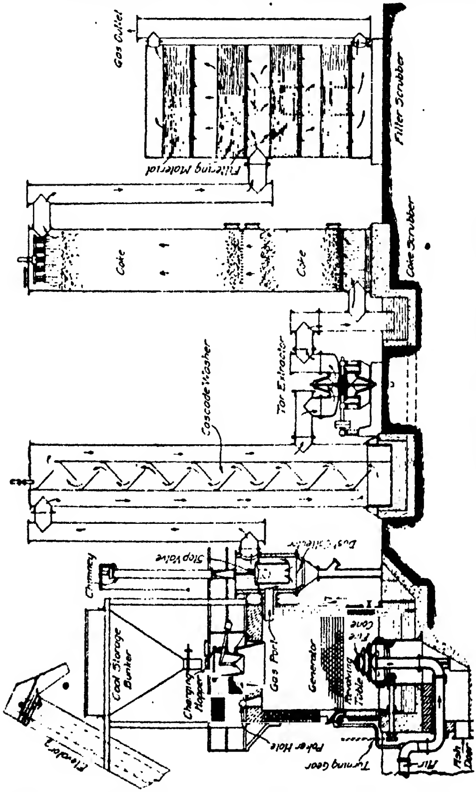


Fig. 19. Section of Crossley Pressure Bituminous Plant. 1000 B.H.P.
Courtesy of Crossley Gas Engine Company, Manchester, England

billets to rolling mills, is shown in Fig. 18. This producer is also used for power purposes. The automatic feed consists essentially of a coal tank and a revolving eccentric chute which spreads the

coal out over the surface of the fuel as shown in the illustration. The steam blower is placed in a horizontal position as shown.

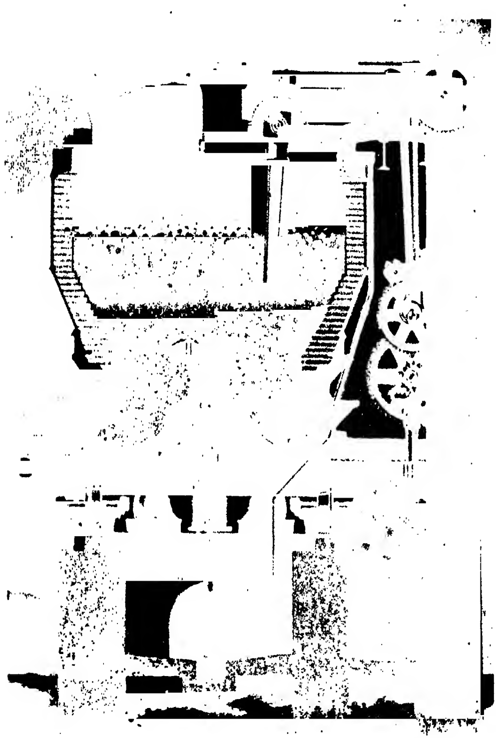


Fig. 20. Section of Standard Type of Hughes Producer
Courtesy of Wellman-Seaver-Morgan Company, Cleveland, Ohio

Crossley Pressure Type. The English pressure bituminous gas plant shown in Fig. 19 includes among its special features, a

revolving table or blast hood, a dust catcher, a cascade or baffle washer, a rotary tar extractor, a coke scrubber, and a wood-wool filter.

Hughes Pressure Type. The producer shown in Fig. 20 has a revolving brick-lined shell with water seals at the top and bottom inclosing the coal to be gasified, an ash support attached to and revolving with it, carrying a blower supplying steam and air, and a water-cooled top plate with a depending vibrating water-cooled poker.

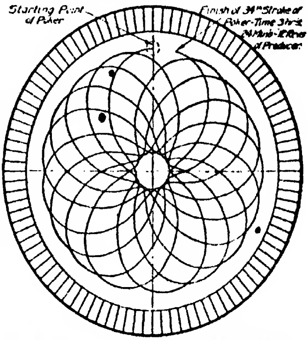


Fig. 21. Horizontal Section of Hughes Producer Showing Path Taken by Water-Cooled Poker, Due to Combined Motions of Poker and Producer

In operation, the incandescent zone of fuel rests upon a bed of ashes extending from the ash pan to a point ranging from 6 inches to 10 inches above the blower hood. The incandescent zone is 10 to 30 inches deep, according to the demand for gas and the consequent condition of the fire. Over this the green coal is spread to a depth of 4 to 8 inches.

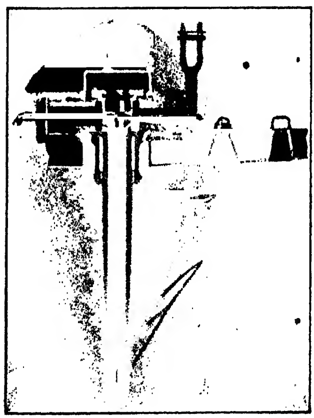


Fig. 22. Section of Water-Cooled Poker
Courtesy of Wellman-Seaver-Morgan Company, Cleveland, Ohio

The water-cooled poker, supported by the stationary top of the producer and extending through the green-coal zone and partly into the

incandescent zone, moves in an arc between the center of the producer and a point within a few inches of the shell lining. As the poker swings backward and forward, the producer shell slowly revolves so that the path in the fuel taken by the poker forms a series of ellipses, as shown in Fig. 21. By this means the bed

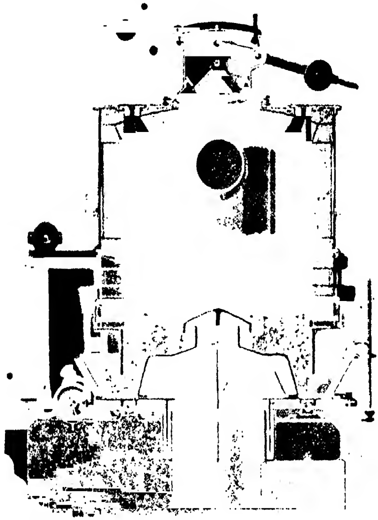


Fig. 23. Vertical Section through Hilger Gas-Producer
Courtesy of The Gas Machinery Company, Cleveland, Ohio

of the fuel is maintained at a constant level and the formation of holes is prevented. A sectional view of the water-cooled poker is given in Fig. 22.

Hilger Pressure Type. The producer shown in Figs. 23 to 27 is one which has been very successful in Europe for steel-mill purposes and is just being introduced into this country. The grate proper consists of two parts; the lower part forms the ash pan and

supports the upper part, a star-shaped distributing hood, shown in Figs. 23 and 24, through which the air and steam are introduced to the fuel bed. The ash pan with the distributing hood is first rotated in one direction for a desired distance and then back again for a somewhat smaller distance; this reversal produces an oscillating motion, and causes the fuel bed to be constantly agitated without discharging more than the desired amount of ashes. The constant

Fig. 24. Horizontal Section through Hilger Gas-Producer
Courtesy of The Gas Machinery Company, Cleveland, Ohio

agitation prevents the formation of large clinker and black spots, at the same time keeping the fire open for the proper introduction and distribution of the air and steam. The mechanism for actuating the rotating grate is shown in Fig. 25. The grate is oscillated by a worm gear, driven by an eccentric rod through a counterweighted dog which flaps over by the motion of the lever. The ashes are automatically discharged by a scraper as shown in Fig. 26. The fuel-charging hopper and method of distribution are shown in Fig. 27. When the valve is open only a little way, the fuel is thrown

toward the center of the producer. By opening the valve wide, the fuel clears the side angles and is thrown to the sides of the producer.

Chapman Rotary Type. The producer shown in Figs. 28 and 29 has the grate fixed (a water-sealed ash pit supporting the fuel bed), and the producer shell is in two parts rotating at different

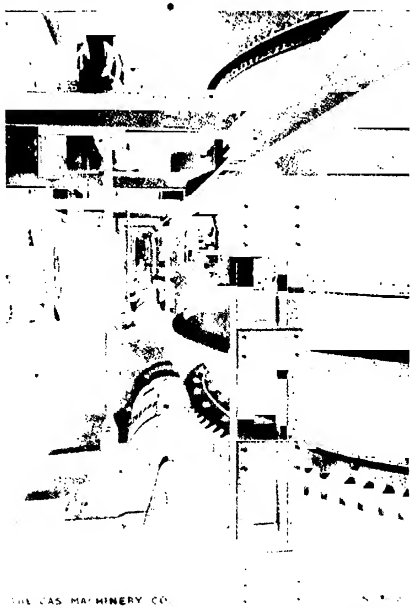


Fig. 25. View of Installation of Twenty Hilger Gas-Producers at Kolpino, Russia
Courtesy of The Gas Machinery Company, Cleveland, Ohio

speeds, the upper part moving faster than the lower. The top of the producer is stationary and supported on three columns; it is in the form of a water tray to keep the top of the producer cool, and supports the charging hopper, fuel chamber, and gas outlet. The revolving sections are supported by rollers, mounted in pairs on

equalizing yokes, the side thrust being taken by separate rollers, also mounted in pairs on equalizing yokes. The upper revolving section is brick-lined and is sealed at the top by a water tray formed in the shell into which projects a ring which is an integral part of the top casting. The space between the upper and lower revolving sections

Fig. 26. Rotary Ash Pan of Hilger Gas-Producer
Courtesy of The Gas Machinery Company, Cleveland, Ohio

is sealed in a similar manner. The lower revolving section is water-jacketed and is sealed at the bottom by the water-sealed ash pit. The upper and lower sections are revolved continuously in the same direction, but at different speeds, by means of gears. This causes the upper portion of the fire bed to revolve over the lower, thus pro-

ducing a shearing action which prevents the formation of holes in the fire. The charging hopper discharges into a stationary fuel chamber, the bottom of which is water-cooled, and as the producer

Fig. 27. Fuel-Charging Hopper of Hilger Gas-Producer
Courtesy of The Gas Machinery Company, Cleveland, Ohio

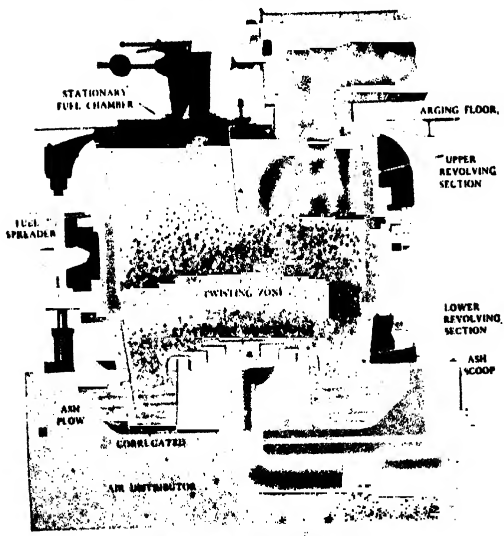


Fig. 28. Chapman Rotary Gas-Producer
Courtesy of Chapman Engineering Company, Mount Vernon, Ohio

shell revolves, the rounded tip of this fuel chamber spreads and packs down the top of the fuel bed. Any inequalities or holes in the top of the fuel bed are filled by gravity by the coal contained in the fuel chamber.

The inside wall of the lower revolving section and the outer edge of the blast hood are corrugated and, since the blast hood is

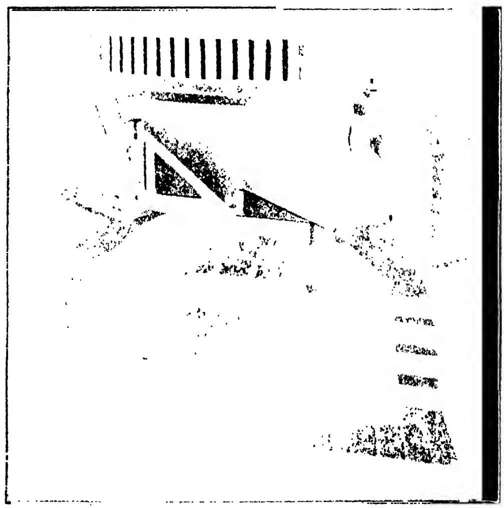


Fig. 29. Lower Portion of Chapman Gas-Producer, Showing Ash Ejector in Operation
Courtesy of Chapman Engineering Company, Mount Vernon, Ohio

stationary, serve to crush the large clinkers and ash. The ashes, after being crushed, are forced up to the surface of the ash pan by three blades, or plows, attached to the lower revolving section. As fast as the ashes come to the surface of the pan, they are picked up by the ash scoops which are also attached to the lower revolving section, and carried to the point from which they are automatically swept into an ash pit or car, as the case may be, as in Fig. 29.

The blast is produced by a five-stage steam-jet blower which, together with the method of regulation, is shown in Fig. 30. This producer is made only in one size (10 feet inside diameter) and requires from $1\frac{1}{2}$ to $2\frac{1}{2}$ horsepower to operate, although 5 horsepower is sometimes required to start.

Blue Water-Gas Generator. The producer shown in Fig. 31 is an intermittent blue water-gas generator. The gas plant consists of a generator, lined with fire brick, and having gas connections at top and bottom, leading into a vertical main. This main terminates at the top in a stack, and at the bottom is connected to the scrubber. Between the generator and this vertical main are located the special gas control valves, which are arranged to operate from the charging platform. Near the bottom of the generator is located a blower connection, fitted with a quick-opening valve, arranged to operate from the charging platform. The blower operates on high pressure and furnishes an excess volume of air, thus shortening the time of blast and decreasing the losses. Steam connections are made below the ash pit, and above the fire line, by means of diverging expansion nozzles, fitted with steam separators and traps, thus allowing of thorough drying of the steam before it comes into contact with the hot fuel bed, resulting in a gain in efficiency.

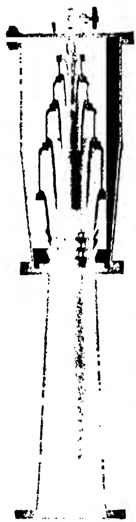


Fig. 30. Steam-Jet Blower

Courtesy of Eynon-Evans Manufacturing Company, Philadelphia, Pa.

Fire is kindled in the generator in the usual manner, with wood or other combustible material, and a layer of fuel is then charged in, and ignited. The blower valve is then opened, the upper gas valve and stack valve being open, and, with the charging door closed, the ignited material in the generator is blasted to an incandescent condition. The blower is now shut off, and the stack valve closed. The steam is admitted, passes up through the bed of hot fuel, and out at the top as water gas, going over through the top connection into the vertical main, and down into the bottom of the wet scrubber.

In a short time, the fire becomes so cool that no more gas is produced, whereupon the stack damper is opened, and the air blast is again turned into the ash pit and the fire brought to incandescence. Then the blast valve, top connection, and stack valve are closed and steam is admitted above the fuel bed, through the upper connection, and comes out at the bottom as water gas, passes into the

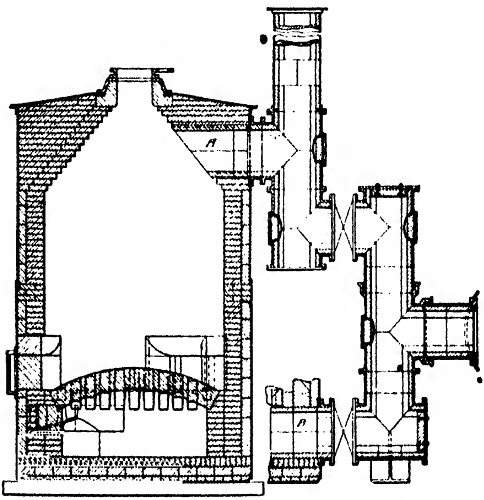


Fig. 31. Cross-Section of Blue Water-Gas Generator
Courtesy of Power and Mining Machinery Company, Milwaukee, Wisconsin

vertical main, and so through the scrubbers to the holder. The object of this reversal of steam flow is to hold the fire at the bottom of the bed of fuel.

From time to time, between intervals of gas-making, fuel is charged through the charging door at the top of generator.

BALANCED-DRAFT TYPE

Both Suction and Pressure Used. Any pressure producer can be operated as a balanced-draft producer by installing an exhauster,

or exhaust fan, between the scrubber and gas holder—or engine, if no holder is used. Some types of the large suction producers could also be adapted to this principle by the installation of a pressure blower at the air inlet to the producer. Up to the present time the pressure type of producer fitted with a blast hood for the distribution of the air and steam blast is the only one to which this principle has been applied.

SPECIAL TYPES

By-Product Gas-Producers. *General Characteristics of Methods.* All by-product processes differ in detail only. They all are based on the same fundamental points; namely, a cooling of the gas after it leaves the producer, washing, and treatment with some reagent to precipitate the by-product.

Ammonium sulphate is about the only by-product that has enough commercial value to justify the additional expense required to save it, and its principal use is that of a fertilizer for certain soils. The sulphate of ammonia is formed from the ammonia in the gas. Nearly all coal contains some nitrogen, usually about 1.5 per cent. From one-tenth to two-tenths of the nitrogen in the coal will go into the gas in the form of ammonia. By the use of an excessive amount of steam the yield of ammonia may be increased very much.

The gas-producer is usually of the pressure type and generally very little different from other producers. The by-product features are introduced after the gas leaves the producer proper. The scrubbing system must always be large and complicated; the by-product system is not adapted for small plants, and the additional first cost of the apparatus necessitates a large outlay of money. The operating expenses will also be higher on account of the salary of a skilled chemist required to handle the plant, reagents for the process and laboratory, and the necessity of advertising the by-product. To make the plant a profitable investment, the revenue from the by-product must be a considerable amount.

Mond System. The Mond by-product system is the only one that has been used to any extent in this country, and it will now be described and illustrated. A diagrammatic section is shown in Fig. 32; and Fig. 33 shows the producer, regenerator, and gas washer. Referring to Fig. 32, *A* is the producer with water-sealed ash pan *B*. *C* is the air regenerator; the hot gas from the producer is passed

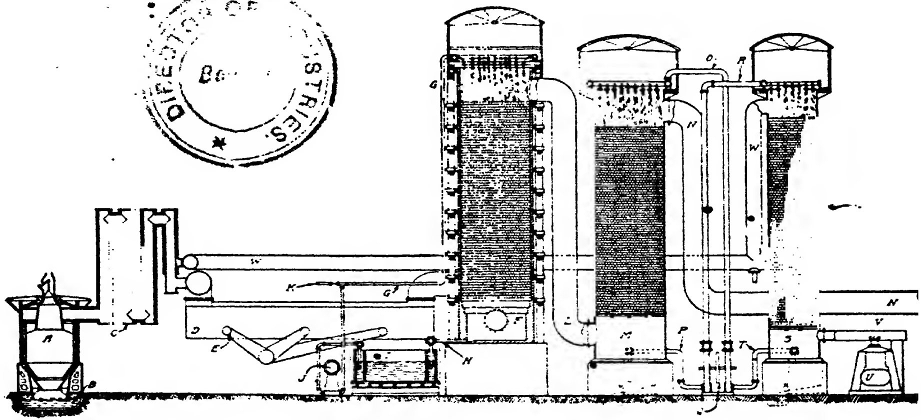


Fig 32. Diagrammatic Section of Mond By-Product Gas-Producer

through this and serves to pre-heat the incoming air, which passes through the outer compartment of the regenerator. *D* is a mechanical gas washer. A few inches of water are in the bottom, and as the gas passes through, the rotating paddles, or dashers, *E* throw the water upward and secure a thorough mixture of the gas and water. In this way a large number of the impurities are washed out. From

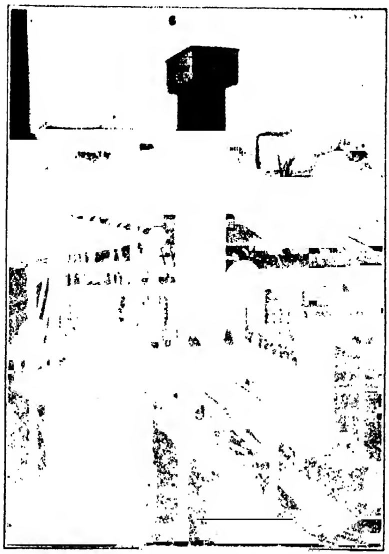


Fig. 33. Producer, Regenerator, and Gas Washer of Mond By-Product Plant

D the gas goes to the bottom of the acid tower *F*. This tower is filled with checkerwork, and diluted sulphuric acid is introduced at top by the pipe *G*. As the gas goes upward, it is brought into intimate contact with the acid and this acts on the small percentage of ammonia in the gas, forming ammonium sulphate. This sulphate of ammonia solution collects at the bottom of *F* and then drains to

the tank *I* by means of pipe *H*. *J* is a circulating pump which takes the liquor from *I* and delivers it to the top of *F* by pipe *G*. The liquor is circulated in this way until it reaches a certain degree of saturation; then some of it is by-passed out of the system by pipe *K*, and a corresponding amount of fresh acid added to the tank *I*. The concentrated ammonium sulphate solution is then evaporated and the sulphate reduced to a solid crystalline state. From the top of tower *F* the gas goes to the bottom of the cooling tower *M* by pipe *L*, and then goes up and out through pipe *N*. *O* is a pipe delivering cold water to the top of *M*. As this water trickles down through *M* it becomes heated by absorbing the heat from the ascending gas. The hot water from the bottom of *M* is withdrawn by pipe *P* and double circulating pump *Q*, and then delivered to the top of the air-heating tower *S*. *U* is an air blower that furnishes the air to the producer *A*. *V* is a pipe connecting *U* with the bottom of *S*. As the cold air goes up through *S*, it becomes heated and saturated by the hot water from *R*. From the top of *S* the air goes to the regenerator *C* by means of pipe *W*. The cold water collecting at the bottom of *S* is withdrawn by pipe *T* and the double-circulating pump *Q*, and delivered to the top of *M*. From the description just given it will be seen that the water acts as a heat carrier between the gas-cooling tower *M* and the air-heating tower *S*.

PRODUCER DETAILS

The various methods of charging the fuel, sealing the charging hopper, top and bottom, to prevent the escape of gas while filling the hopper and charging the fuel, distribution of the blast by means of blast hoods, and location of gas outlets, types of grates, sealing of poke- and peek-holes, etc., have been shown in the preceding illustrations.

Fire-Brick Lining. The design and usual method of installation of the fire-brick lining is shown in Fig. 34. A space of about an inch is left between the inside of the shell and the outside of the fire brick and is filled with some elastic, fire-resisting material in order to allow for the unequal expansion of the fire-brick lining and the producer shell and to prevent the leakage of air from the ash pit to the top of the fuel bed through the space in back of the bricks.

Regulation of Steam Supply. Any gas-producer to be operated efficiently must be supplied with the proper amount of steam, and furthermore no more steam must be delivered to the producer than it is able to decompose. If an excessive amount of steam is used, it will pass through the fire without being decomposed, will chill the fire, and add water vapor to the gas. In some cases the chilling effect may be enough to stop the process of gasification. If not enough steam is used, the fire will become hotter and the producer efficiency will be reduced. In a power gas-producer the quantity of gas made should vary with the load on the engine. As the latter may vary in some cases from engine friction up to full load, it is evident that the rate of gasification must also vary through a large range. As a result of the conditions just mentioned, extreme care will be necessary in proportioning the amount of steam delivered to the producer to the amount of gas made therein. Several devices intended to accomplish this regulation are illustrated and described below. There are two types of such apparatus. In the first type, the amount of steam delivered to the producer is regulated by the varying pressure or suction on the producer. In the second type, advantage is taken of the fact that the best results in the producer are obtained when the amount of steam delivered to the producer is that which the air blast will take up when heated to the neighborhood of 150 degrees F. Consequently, the air blast is kept heated to that temperature by means of a thermostat.

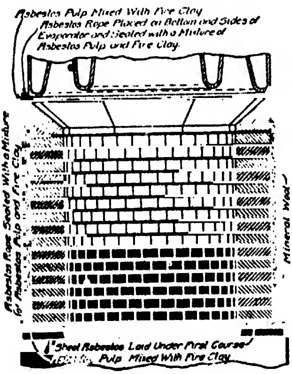


Fig. 34. Typical Section Showing Fire-Brick Lining for Gas-Producer
Courtesy of Otto Gas Engine Works, Philadelphia, Pennsylvania

Smith Suction Regulator. The regulator shown in Fig. 35 is of the first type and is applied to suction producers of both types,

although lately suction producers of large size of this make, in which the suction is furnished by an exhauster, have been fitted with thermostatic regulation in place of this type of regulator. This figure also shows the heater for vaporizing the water and superheating the resulting steam and pre-heating the air by utilizing the waste heat in gas-engine exhaust gases. *A* is the inlet for the exhaust gases while *B* is the interior of the heater. *C* is a thin, flat disk, around which the exhaust gases circulate and through which the air and steam

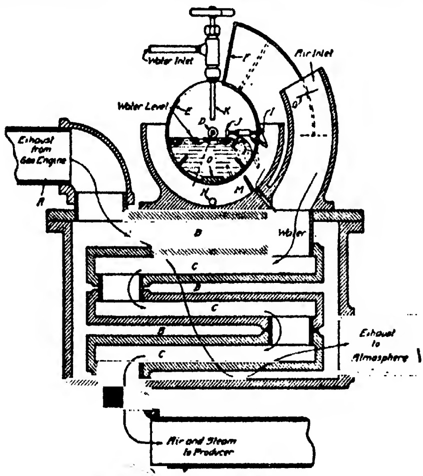


Fig. 35. Section of Smith Suction Regulator

pass. *D* is a shaft upon which the weighing vessel *E* is pivotally supported. *F* is a rod connecting *E* with the vane *G*. The air inlet is curved with its center at *D*. *J* is a screw for adjusting the amount of water going through the orifice *I*. *K* is the water inlet pipe, and is controlled by a valve. If more water is delivered to *E* than can pass out through *I*, the excess is drained to *M* by an opening not shown in the figure and then passes out through the drain *N*. *O* is a counterweight to keep *E* poised in the position shown in the illustration. The operation is as follows:

When the engine draws gas from the producer, outside air will rush in through the air inlet to replace the gas removed. As it goes in past G it will cause this vane to be deflected and take the position indicated by the dotted lines; at the same time, E will be moved a corresponding amount and water will pass out of I and go down to the vaporizer below. When the sucking action of the gas engine ceases, the flow of air in H will cease and the counterweight will swing E and G back to their normal position; just as soon as I comes back to this position the flow of water will stop. The water falling down on the hot disks C is converted into steam and swept on through by the movement of the incoming air. In this passage the air becomes pre-heated and the steam superheated. At the next charging stroke of the engine the same cycle will be repeated. The amount of water delivered each time may be adjusted by means of the screw J.

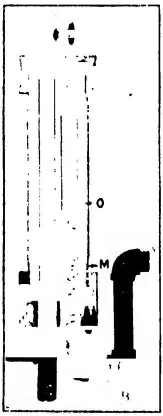


Fig 36. Section of Syracuse Water Regulator

Courtesy of Syracuse Industrial Gas Company, Syracuse, New York

Syracuse Regulator. The regulator shown in Fig. 36 supplies water to the vaporizer in proportion to the load on the engine. There are no moving parts to get out of order or adjustment. Pipe B supplies water and pipe A carries off the excess not taken by the regulator. Pipe C is connected to the engine gas-supply pipe and pipe D leads to the producer.

When the engine is idle, the water stands at M but when the engine is drawing gas it rises, due to the suction, to some point O. The water runs out of the slot shown and through pipe D to the producer. The amount of this water is proportional to the load on the engine.

The regulator shown in Fig. 9 is of the same general type as that in Fig. 36.

Fairbanks-Morse Pressure Regulator. Fig. 37 is a diagrammatic view of a regulator in which $A\,F$ is a glass U-shaped tube having a branch at B. C is a tank containing water maintained at a constant, predetermined level. D is a glass tube having its lower end ground

off at a more or less acute angle and the opening thus formed is submerged just below the surface of the mercury contained in the lower part of the U-tube. The leg *F* at *E* of the U-tube is connected with the gas main. The operation is as follows: As the pressure of gas in the main varies, due to varying demand for gas, so will the relative levels of the mercury change, i.e., as the demand for gas increases, the mercury will rise in the leg *F* of the U-tube and descend in the other leg, uncovering more of the angular end of the tube *D* and thus increasing the area of the opening and allowing more water to flow from the tank *C* to *B* where it falls into the cup *G* and runs to the vaporizer.

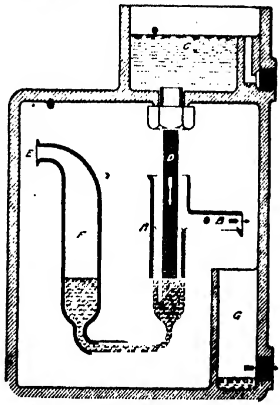


Fig. 37. Cross-Section Showing Principle of Operation of Automatic Water Regulator for Fairbanks-Morse Suction Gas-Producer
Courtesy of Fairbanks, Morse, and Company, Chicago, Illinois

The apparatus is mounted in a cast-iron box with a glass front, so that it is protected against breakage or from any interference, but is at all times visible.

Smith Thermostatic Regulator. A thermostatic regulator is shown in Fig. 38. In this the thermostat *A* is placed in the air inlet

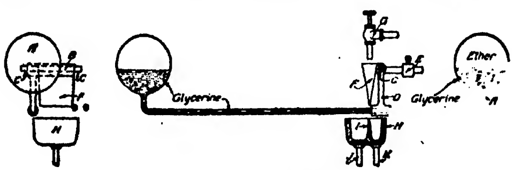


Fig. 38. Smith Thermostatic Water Regulator

to the producer and consists of two copper balls connected by a drawn-copper tube. This thermostat swings on knife-edges *C* formed on the rod *B* which supports the thermostat through the

bracket D. The bracket is also provided with an arm on which is mounted a sliding counterweight E. The balls are partially filled with glycerine. The remaining space in one of the balls, on top of the glycerine, is filled with ether. The thermostat is balanced on the knife-edges at atmospheric pressure by means of the counterweight E. As the temperature of the air passing the thermostat increases, ether is vaporized, displacing the glycerine in that ball and driving it into the other, thus upsetting the balance of the thermostat. The decrease in the amount of water is obtained by a funnel F attached to the knife-edge rod B, receiving water through the valve G. Underneath the funnel is placed a box H which is divided into two compartments by a knife-edged partition I. The box is so located that the partition is directly underneath the funnel when the thermostat is balanced. From one of the compartments a pipe J leads to the vaporizer and from the other a pipe K leads to the water seal or waste pipe. By means of this apparatus the water is diverted from the vaporizer into the waste pipe as the air is heated, until, when the air is too hot, all the water is sent to waste. As the amount of water going to the vaporizer diminishes, the air becomes less heated in the vaporizer. By means of the hand valve G and counterweight E, the thermostat can be set to maintain a constant temperature of inlet air to the producer throughout the range of load of the plant, and by means of the valve G the constant temperature may be varied to a considerable extent.

GAS-CLEANING

Producer gas, in addition to containing condensible constituents as shown in Table I, generally carries fine dust with it. Gas-cleaning is used synonymously with gas-scrubbing and gas-washing and means either the removal of foreign constituents from the gas, or the removal of certain elements of the gas composition that are undesirable for certain uses of the gas. Water is nearly always used in gas-cleaning processes. The object in cleaning any particular gas is simply to prepare it for some special kind of work. No general rules can be laid down for the constituents that must be removed or the degree of purity required. The primary requisite is that the gas shall be adapted to its specific work. The highest degree of purity is required for engine work; at the

same time the additional cost of cleaning the gas up to that point might prohibit its use for heating a furnace where some impurities would not have a detrimental effect.

Gases may be cleaned by means of deflectors, static scrubbers, filters, and mechanical scrubbers.

Deflectors. The deflector consists of an obstruction placed across the path of moving gas, and causes a sudden change of direction of flow. This has a tendency to precipitate the fine dust or water globules carried in suspension. It is very similar to the steam separator used in steam pipes to separate the water from the steam.

Static or Tower Scrubbers. Static or tower scrubbers are those in which the cleaning is done by water in a tower. This class of gas-cleaning apparatus is divided into the following types: bubbling; impinging; hurdle; rain; and baffle.

Bubbling Scrubbers. Bubbling scrubbers are those in which the gas is forced down into a seal of water and then bubbles up through the water. *Impinging scrubbers* are those in which the gas strikes the surface of a water seal at right angles to the direction of flow of the gas at high velocity and is taken off in the opposite direction. These two types of scrubbers are little used.

Hurdle Scrubbers. Hurdle scrubbers are those in which the gas is introduced at the bottom, and water sprayed into the top, of a tower which is filled with coke or layers of wood slats, alternate layers of slats being at right angles to each other. The object of this is to break up the water spray and to bring the gas into more intimate contact with the water. The scrubbers shown in Figs. 5, 10, 11, 12, 14, 16, and 32 are of this type and need no further explanation.

Rain Scrubbers. Rain scrubbers are those in which no hurdles or coke are used. The gas is introduced at the bottom of the tower and in its passage to the top is thoroughly washed by a fine mist of descending water. The water is broken up into a mist in various ways such as spraying by means of nozzles, breaking the water up by means of revolving screens, etc.

Baffle Scrubbers. Baffle scrubbers are small tower scrubbers in which the water is caused to flow from baffle to baffle in a thin film. In this way the gas is forced to follow a zigzag course through film

after film of water and cannot channel and escape unwashed. Baffle scrubbers are shown in Figs. 6 and 19.

Mechanical Scrubbers. The removal of *tar* from gas is one of the hardest problems in connection with gas cleaning. The use of a tar-laden gas in gas engines will quickly gum the valves and necessitate stopping the producer and engine. This is the reason why so many gas-producers for power purposes are using anthracite coal. This particular fuel, producing a gas practically free from tar, makes the problem of gas-cleaning an easy matter. However, there are many cases where the high cost of anthracite coal prohibits its use for producer gas for power purposes. In some places the cost of anthracite coal is four times the cost of bituminous coal, and, since a pound of the latter will make on the average as much producer gas as a pound of the former, there would evidently be a decided advantage in using bituminous coal in a producer-gas power plant. The problem of the use of bituminous coal for such work is simply the problem of eliminating the tar. This may be done by the separation of the tar from the gas or the use of a device that will prevent the formation of the tar.

Tar. Tar is one of the products of the destructive distillation of coal and it is a very complex substance, made up of about two hundred compounds, some of them so complex and hard to isolate that very little is known about them. The exact composition will depend on a large number of factors, the most important of which is the temperature at which it is formed. Coal tar has a specific gravity of about 1.15, a black color, and a very marked and disagreeable odor. It condenses easily and if brought into intimate contact with incandescent carbon it may be converted into fixed gases. The fact just mentioned forms the basis of all tar-destruction methods; that is, where the tar is broken up in the producer.

The separation of the tar from the gas may be accomplished by an extensive tower scrubber or by the use of some form of centrifugal apparatus which will drive the tar out of the gas by centrifugal force. The latter method can be made fairly effective, but the former is adapted only for gas containing small amounts of tar. On the other hand, the centrifugal method requires close watching, and, in some cases, considerable power to run the apparatus; neither method is as satisfactory as the complete destruction of the tar in the producer;

if this is accomplished, the inevitable lampblack, formed by the process, introduces cleaning complications of its own.

Centrifugal Tar Extractors. Centrifugal tar extractors consist essentially of a disk, on either side of which is located a fan, running in a casing into which a spray of water is introduced. This spray forms a seal at the circumference of the casing which seals off one side of the disk from the other. The cleaning action is secured by combination of the cleaning effect of the mist wetting down and enveloping small particles of impurities, and centrifugal force throwing the impurities into the seal from which they are washed into a pit through a dust leg in the bottom of the casing. The bubbling of the gas through the seal also has a cleaning effect.

Theisen Washer. The Theisen gas washer is described on page 129, Gas and Oil Engines, Part II.

Disintegrator Washer. A type of mechanical washer more efficient than either of the previous two is the disintegrator washer. With this apparatus, less power and water are required and the gas is cleaned to a greater degree of fineness. This apparatus consists of a series of cages of rods, blades, or vanes, the alternate cages rotating in opposite directions. The gas is introduced at the circumference and is taken off at the center of the casing. The water is sprayed into the center of the revolving cages and is thrown from cage to cage towards the circumference of the casing, and, being thrown from a cage rotating in one direction on to a cage rapidly rotating in the opposite direction, is broken up into an exceedingly fine mist which is distributed throughout the casing so that every particle of the gas is washed.

Every type of gas-cleaning apparatus in which water is used as the cleaning agent serves also to cool the hot gases. Tower washers employ more water in proportion to the amount of gas cleaned, and are therefore more effective as coolers than mechanical washers.

Smith Spun-Glass Tar Extractor and Gas Cleaner. The apparatus shown in Fig. 39 is a recent development in gas-cleaning. With this type of apparatus the standard anthracite producer is used with bituminous coal and no attempt is made to fix the resulting tar. The raw gas, on leaving the producer, is first cooled to a point where tar vapors are condensed, by being passed through a primary cooler or condenser, from which the gas is carried into a rotary gas

pump or exhauster *B* which delivers it under pressure into the main *C*. It is then delivered through a porous diaphragm of spun glass *E* into the engine main *F* where a sump or separator *G* is provided in which the tar accumulates. The diaphragm must be sufficiently porous to permit the gas and tar to pass freely and is in the form of a uniform layer of glass wool retained between two metal screens. Ordinarily, the thickness is about one-quarter of an inch and the diameter must be adjusted in accordance with the quantity of gas to be treated—ordinarily four hundred cubic feet per hour can be

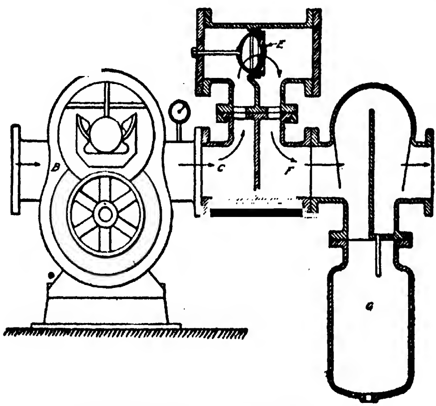


Fig. 39. Section of Smith Tar Extractor and Gas Cleaner
From Transactions of American Society of Mechanical Engineers

handled per square inch of diaphragm area. No tar is retained in the diaphragm, both tar and gas being discharged together, but in passing through, an important change in the physical state of the tar occurs. On the entering side, the tar exists in the form of a large number of minute particles, known as tar fog, while in passing the diaphragm these particles are caused to coalesce so that on the discharge side the tar particles are of relatively large dimensions—so large, in fact, that they can no longer be carried forward in the gas current and immediately separate out by gravity and drain into

the sump. It appears to be possible to secure any desired degree of gas-cleanness simply by regulating the velocity of flow through the diaphragm, i.e., the pressure maintained across the diaphragm. In ordinary commercial operation, it is found that a difference in pressure of from 2.5 to 4 pounds per square inch will clean the gas to such an extent that no discoloration will be produced on a white filter paper through which 30 cubic feet of gas has been passed.

This is not a filtration process, since, for the best separation by filtration, the velocity must be low and the material separated out remains in the filter. No water is used in connection with this process except that required to cool the gas, and as a consequence there is no formation of tar emulsion—therefore the tar separated is practically water-free (less than 1 per cent) and may easily be burned. The gas must be cooled only sufficiently to completely condense the tar vapors, since any further cooling will increase the viscosity of the tar and consequently the resistance through the diaphragm—which must be a minimum. This process is not well adapted for use on gas containing large quantities of lampblack or from coals yielding a very heavy, viscous tar. It has, however, been used with great success with Ohio, Illinois, and Indiana high-volatile coals, and with lignite.

The theory of the operation of this extractor is not definitely known, but it is supposed that it is the result of some electrical action caused by the impact and fluid friction against the glass wool.

This apparatus has also been used with marked success for cleaning gas made from anthracite coal, giving a much cleaner gas with a lower water consumption than can be obtained by other methods.

PRODUCER-GAS PLANTS

Comparison between Producer Gas and Steam. The high fuel and water economy of the producer-gas power plant is one of its strongest advantages over the steam-power plant. The results of the comparative tests on a producer plant of about 200 electrical horsepower, and on a steam plant with a boiler of about 200 boiler horsepower and non-condensing engine, made at the United States Geological Survey Coal Testing Plant, are given in Table III. It should be noted, however, that a condensing steam engine would have used 30 to 40 per cent less coal than the non-condensing engine.

TABLE III

U. S. Government Comparative Test of Steam and Producer Plants

Name of Coal	Water Evaporated from and at 212° F. per pound Dry Coal Steam Plant	Total Dry Coal per Electrical H. P. per Hour *	
		Steam Plant	Gas-Producer Plant
Alabama No. 2	8.55	4.08	1.64
Colorado No. 1	7.21	4.84	1.71
Illinois No. 3	8.04	4.34	1.79†
Illinois No. 4	7.27	4.80	1.76†
Indiana No. 1	8.45	4.13	1.93†
Indiana No. 2	8.02	4.35	1.55†
Indian Territory No. 1	8.64	4.04	1.83
Kentucky No. 3	8.27	4.22	1.99†
Missouri No. 2	7.08	4.93	1.71†
West Virginia No. 1	8.95	3.90	1.57
West Virginia No. 4	9.65	3.62	1.29
West Virginia No. 9	10.09	3.46	1.59
West Virginia No. 12	9.90	3.53	1.50†
Wyoming No. 2	5.92	5.90	2.07

* In gas-producer plants, this includes the coal consumed in the producer and the coal equivalent of the steam used in operating the producer.

† Gas-producer hopper leaked during these tests.

There is little difference in the coal or water consumption between large and small producer-gas plants. This is due to the fact that small gas engines may have practically the same thermal efficiency as large ones. The small producer-gas power plant can be operated nearly as cheaply as a large plant, so that it is not necessary to use large units in order to get economical results. In many cases where the load fluctuation is large, much better results will be obtained by installing, say, four 500-horsepower gas-engine units in place of one 2000-horsepower unit. Even a small producer-gas plant is more economical of fuel and water than a large steam plant. The economy of water of the producer-gas plant over the steam plant in cases where water is scarce or impure, so as to make it undesirable for boiler use, is of the greatest importance. The gas-producer does not make any smoke, so that the producer-gas power plant offers a solution for the smoke problem.

The labor in a producer-gas plant will generally be about the same as that in a similar steam plant, and it is easier to install mechanical appliances for handling the fuel in a gas plant than in a steam

plant. The producer may be started after standing-by, in about twenty minutes, and can be stopped instantly.

The first cost and cost of repairs will be about the same in producer-gas as in steam plants. Gas engines cost slightly more per horsepower than steam engines, but the cost of the smokestack is eliminated. In small producer-gas plants the cost is about one-fifth higher than in steam plants.

In steam and producer-gas plants the steam and producer gas simply act as carriers of thermal energy. The cooling of the steam will lower, and may entirely destroy, its thermal energy, while the cooling of the gas will simply decrease its volume and increase the thermal energy carried per cubic foot. This last statement refers to calorific power, only; since the sensible heat of the gas is of no use in the gas engine, the temperature of the gas as it leaves the producer should be very low. In other words, with producer gas the thermal energy carried by the gas *for* the gas engine will *not* be lowered if the gas is cooled. This fact makes it possible to put in central producer-gas plants with long pipe lines to distribute the gas to isolated engines. It is consequently possible to build a large producer-gas plant at the coal mines and, in place of shipping the coal to the various places of consumption, to pipe the gas to those places.

The use of a gas-holder for storing the gas has marked advantages in certain cases. By this means, irregularities in the load may be taken care of without any difficulty. In some industries it may be desirable to have a small amount of gaseous fuel for heating furnaces—forges, for example—and in such cases the gas may be taken from the same holder that supplies the engine.

Fig. 40 shows a typical producer-gas power plant. The holder is frequently placed farther away from the producer than shown in the illustration; in fact, it may be placed on any area that is available, regardless of immediate proximity to the producer.

Growth of Industry. The charts of Fig. 41 show the increase in the use of producer-gas power in the United States in the decade 1900-1911, both in number of plants and in the horsepower installed.

In March, 1912, there were between 950 and 1000 producer-gas power plants in the United States, ranging in size from fifteen horsepower to several thousand horsepower. Statistical data from these

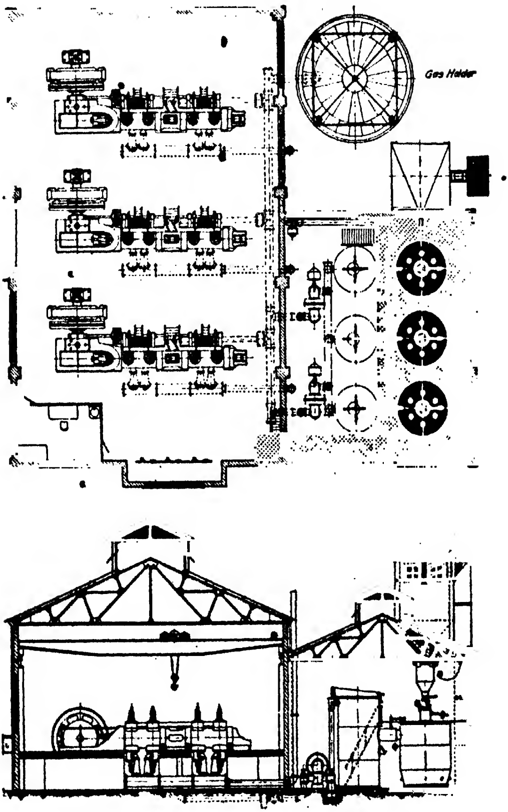


Fig. 40. Plan and Elevation of 1500-Horsepower Pressure Producer Plant for Bituminous Coal
Courtesy of R. D. Wood and Company, Philadelphia, Pennsylvania

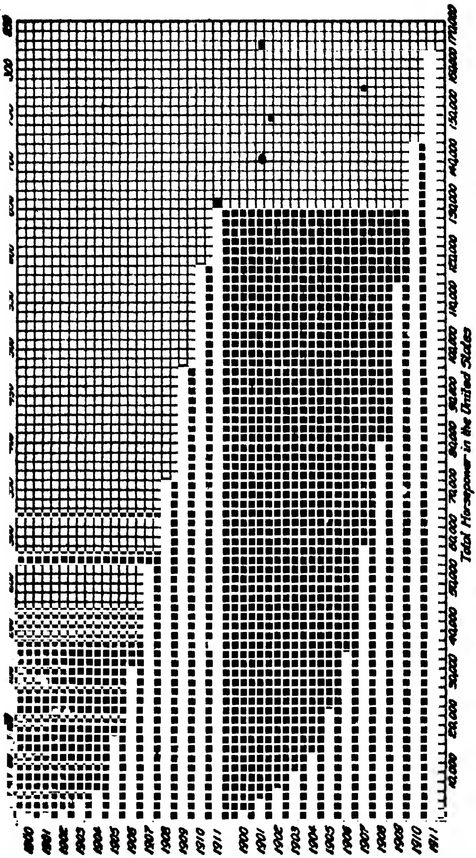


Fig. 41. Installations and Horsepower of Producer-Gas Plants in the United States 1900 to 1911

TABLE IV

Summarized Data Relative to Producer-Gas Power Plants in the United States, March, 1912

Fuel	Number of Plants			Horsepower									Percentage of Total Number		Percentage of Total Horsepower	
				Total			Average		Minimum		Maximum					
	1909	1912	Percentage Increase	1909	1912	Percentage Increase	1909	1912	1909	1912	1909	1912	1909	1912	1909	1912
Anthracite coal:																
Over 500 h. p.	8	29	263	7,550	25,825	242	959	890	600	520	1,500	1,800				
500 h. p. or less	407	581	43	40,550	63,615	57	100	110	15	15	500	500				
	415	610	47	48,100	89,170	86	116	146	15	15	1,500	1,800	88	84.5	43	47.8
Bituminous coal:																
Over 500 h. p.	20	40	100	49,000	76,890	57	2,450	1,920	750	600	6,000	9,000				
500 h. p. or less	17	37	118	5,150	9,725	89	300	262	35	25	500	500				
	37	77	108	54,150	86,615	60	1,460	1,120	35	25	6,000	9,000	8	10.7	49	46.3
Lignite:																
Over 500 h. p.	3	3	0	7,275	7,275	0	2,430	2,430	525	525	3,750	3,750				
500 h. p. or less	19	29	53	1,725	2,955	71	90	102	25	25	250	350				
	22	32	45	9,000	10,230	14	410	320	25	25	3,750	3,750	4	4.4	8	5.1
Wood:																
Over 500 h. p.	0	0		0	0											
500 h. p. or less		1		.	500	..		500	..	500	...	500	....	0.1	...	0.[illegible]
Oil:																
Over 500 h. p.	0	0	.	0	0											
500 h. p. or less	..	2		....	325	.	.	103	..	100	.	225	.	0.3	..	0.2
Total	474	722	52	111,250	187,140	68	235	260	15	15	6,000	9,000	100	100	100	100

plants are given in Table IV, which classifies them according to their size and the kind of fuel used.

Special Uses of Producer Gas. Producer gas has been used to a limited extent for firing steam boilers, in Europe. It is not very good engineering practice and should be used only in cases where steam is required in the process of manufacture; as, for instance, in steaming lumber. It will be much better to eliminate the boiler entirely in other cases and to use the gas directly in a gas engine. In general, no direct fuel economy will result from first gasifying the fuel in a gas-producer and then burning the resulting gas under a steam boiler.

Firing Ceramic Kilns. Producer gas has been used extensively in Europe for firing ceramic kilns, but until the last few years has had a very limited use for this purpose in America. Several costly failures have been made in attempting to use it, but these

have not been the fault of the system but rather were due entirely to the ignorance of the men who have attempted to use it. Producer gas has decided advantages for ceramic work, but great care is necessary in applying it. The best results will be obtained in connection with continuous kilns. The use of producer gas in kilns eliminates clinkering in the kilns, induces more uniform burning, produces better combustion, makes it possible to regulate fire more readily, secures a centralization of furnaces, and should result in fuel economy.

Firing Metallurgical Furnaces. The first, and, even today, the largest, field for the use of producer gas lies in firing metallurgical furnaces. It has been an important factor in developing the steel industry and has become a commercially necessary adjunct of it.

Producer Rating. It is customary to rate the capacity of producers in horsepower, i.e., a producer of a certain capacity will supply an engine of that capacity with gas continuously, or at least over a considerable period of time. To do this, a certain weight of fuel per square foot of grate area must be gasified per hour. This weight, and the consequent quantity of gas generated, will vary greatly with the kind of fuel used, and is at best an uncertain quantity.

In the early stages of development in this country, design followed European practice and brought about a great deal of trouble in meeting guarantees, and in some cases caused the entire failure of the plant. This failure was due to the fact that European builders, in making their guarantees, specified certain high grades of fuel, in which the ash rarely exceeded one and one-half per cent, while American coals contain a much larger amount of easily fusible ash. There is still a tendency to slightly overrate producers, which causes clinkering and other troubles if the producer is forced to its full rating for any length of time, although satisfactory operation may be obtained while the plant is operating below rated load.

The following rates of fuel consumption have been shown by tests and practical operation to be allowable for the various fuels:

Anthracite, 8-10 lb. as fired, per sq. ft. of grate surface, per hour
Bituminous, 5-11 lb. as fired, per sq. ft. of grate surface, per hour
Sub-bituminous, 7.5-12 lb. as fired, per sq. ft. of grate surface, per hour
Lignite, 7-10 lb. as fired, per sq. ft. of grate surface, per hour
Peat, 12 lb. as fired, per sq. ft. of grate surface, per hour

TABLE V

Heat Data, Fuel Composition, and Gas Analysis for Bituminous Coal and Low-Grade Fuels

Item	Bituminous Coal	Sub-bituminous Coal	Lignite	Peat
Number of tests averaged	112	7	7	1
B.t.u. per pound	12,370	9,910	7,110	8,130
Cu. ft. of gas per pound—yield	61.1	39.3	27.7	30.3
Pounds per sq. ft. of fuel bed, per hour	7.64	11.02	13.28	16.2
B.t.u. per cu. ft. of gas—standard	151	150	161	175
Producer (cold) efficiency—per cent	74.6	63.1	62.8	65.2
Pounds of tar, soot, etc., per ton of fuel				
Water not extracted	354	250	157	240
Water extracted	287	224	...	157.
Composition of fuel (per cent)				
Moisture	6.6	15.0	35.7	21.0
Volatile	32.8	34.3	29.2	51.7
Fixed carbon	50.6	39.4	27.2	22.1
Ash	10.0	11.3	7.9	5.2
Sulphur (separately determined)	2.32	0.90	1.12	0.45
Volumetric analysis of gas (per cent)				
Carbon dioxide	9.71	11.16	9.90	12.40
Oxygen	0.02	0.12	0.13	0.00
Ethylene	0.19	0.20	0.10	0.40
Carbon monoxide	19.03	17.52	20.86	21.00
Hydrogen	13.48	14.41	14.30	18.50
Methane	2.78	3.64	2.88	2.20
Nitrogen	54.79	52.95	51.83	45.50

Consumptions of lignite of more than forty pounds per square foot have been reported, but it is unwise to choose so high a figure as this unless the characteristics and action of the particular lignite to be used are very well known.

A moderate rate of driving for all fuels is absolutely essential for continuous operation, since thereby clinkering is reduced to a minimum and, consequently, the producer can be easily cleaned in operation.

RESULTS OF PRODUCER-PLANT TESTS

Producers for the gasification of anthracite coal have become fairly well standardized. Until recently, however, the use of bituminous and low-grade fuels was attended by troubles of various kinds. The progress that has been made in this direction points to the ability soon to use all grades of bituminous coal and low-grade fuels in producer work. The following are the summarized results from the Government tests at St. Louis, Norfolk, and Pittsburgh as averaged and reported by R. H. Fernald.

TABLE VI

Gas Yield of Coals of Different Calorific Values

Approximate Calorific Value B.t.u. per Lb. of Dry Coal	Coal per Sq. Ft. of Fuel Bed per Hour (Pounds)		Gas under Standard Conditions			Producer Efficiency Per Cent
			Per Lb. of Dry Fuel		B.t.u. per Cu. Ft. High Value	
	As Fired	Dry	Calorific Value B.t.u.	Yield Cu. Ft.		
15,000	5.64	5.50	13,350	91.1	153	74.7
14,500	6.10	5.92	12,460	80.5	160	74.2
14,000	6.45	6.22	10,890	72.1	152	71.9
13,500	6.97	6.54	10,070	67.8	150	69.3
13,000	7.88	7.30	9,360	61.8	152	67.5
12,500	8.76	7.84	8,770	58.6	149	65.0
12,000	8.54	7.69	8,780	59.4	148	67.6
11,500	10.11	8.88	8,010	54.8	146	64.3
11,000	11.61	10.53	6,110	45.9	133	52.5
10,500	13.26	11.53	5,790	40.9	135	50.5

Standard Conditions: 62° F. and 14.7 lb. per sq. in.

In Table V all results affected by the load factor are given for loads ranging only from 90 to 100 per cent of full load and no test of less than 30 hours is included in the average. It should be noted that all the results of coal consumption are referred to the weight of the coal as fired. All heat values which have been chosen are higher values.

In Table VI is given a classification of 103 coals condensed into 10 groups, the grouping being made according to calorific value, each group embracing a range of 500 B.t.u., the middle figure of which is assigned as the approximate calorific value, on a dry-coal basis, of the group. The duration of the tests from which the results were obtained range from 29 to 74½ hours, with an average duration of 48 hours.

In Table VII are given typical analyses of producer gases made from the various fuels in the up-draft type of producer, while in Table VIII the same sort of analyses are given for down-draft producers.

The quantity of gas obtained varies with the fuel used, with the type of producer plant, and with the method of operation. In

TABLE VII

Up-Draft Pressure Producer Gas

(Percentages by Volume)

Constituents	From Anthracite Coal	From Bituminous Coal	From Lignite	From Peat
Carbon dioxide..... CO_2	5.2	9.84	10.55	12.40
Oxygen.............. O_2	0.4	0.04	0.16	0.00
Ethylene.......... C_2H_4	....	0.18	0.17	0.40
Carbon monoxide... CO	22.9	18.28	18.72	21.00
Hydrogen............ H_2	15.3	12.90	13.74	18.50
Methane .... CH_4	1.0	3.12	3.44	2.20
Nitrogen............ N_2	55.2	55.64	53.22	45.50
	100.0	100.00	100.00	100.00

Table IX is given the average yield of producer gas in cubic feet and B.t.u. per pound of fuel, as obtained at the government testing plant at St. Louis (with the addition of values for coke or charcoal, and anthracite).

The results obtained from some of the low-grade fuels are summarized in Table X.

In Table XI are given some test results and total plant efficiencies of gas-producers supplying engines, which have been previously reported on in Table XXIX, Gas and Oil Engines, Part III.

TABLE VIII

Down-Draft Producer Gas

(Percentages by Volume)

Constituents	From Bituminous Coal	From Lignite	From Peat
Carbon dioxide................ CO_2	6.22	11.87	10.94
Oxygen .... O_2	0.13	0.01	0.41
Ethylene....................... C_2H_4	0.01	0.00	0.06
Carbon monoxide .... CO	21.05	16.01	16.91
Hydrogen...................... H_2	12.01	14.76	10.19
Methane...................... CH_4	0.49	0.98	0.66
Nitrogen...................... N_2	60.09	56.37	60.83
	100.00	100.00	100.00

TABLE IX

Yield of Producer Gas per Pound of Fuel

Character of Fuel	Yield of Gas Cu. Ft. per Lb. of Fuel		Higher Heat Value of Gas B.t.u. per Cu. Ft.	Higher Heat Value of the Gas B.t.u. per Lb. of Fuel	
	As Fired	Dry		As Fired	Dry
Coke or charcoal..	85	90	140	11,900	12,600
Anthracite coal..	70	75	135	9,450	10,100
Bituminous coal	60	65	152	9,120	9,880
Lignite..........	36	46	158	5,690	7,270
Peat.....	30	38	175	5,250	6,650

TABLE X

Moisture, Ash, and Fuel Consumption for Low-Grade Fuels

Character of Fuel	Moisture Per Cent	Ash Per Cent	Fuel Consumed in the Producer, Lb. as Fired per B.h.p. per Hour
Bituminous—run-of-mine, slack, bone, and washery refuse..................	0.47—16.69	19.63—43.74	1.10—3.98
Peat......................	13.88	27.78	3.37

TABLE XI

Producer Test Results and Total Plant Efficiency of Producers Supplying Gas for Engines Reported in Table XXIX, Gas and Oil Engines, Part III

C*—LOOMIS-PETTIBONE DOWN-DRAFT SUCTION						
Fuel				Producer Efficiency Per Cent	Thermal Brake Efficiency Per Cent	Plant Efficiency B.h.p. Per Cent
Kind	Lower Heat Value B.t.u. per Lb.	Rate of Gasification—Lb. per Sq. Ft. of Fuel Bed Area per Hour	Consumption Lb. per B.h.p. per Hour			
Bituminous run-of-mine	13,500		1.34	62.2	22.6	14.1
E*—LOOMIS-PETTIBONE DOWN-DRAFT SUCTION						
Fuel						
Bituminous Clearfield	14,320	13.36	0.97	73.8	24.95	18.4
Average of 9 weeks' operation. Load factor 72.2%, 14 hours daily stand-by			1.23 1.36	Excluding coke for new fires Including coke for new fires		

*Reference letter, see page 259, Gas and Oil Engines, Part III.

CARE OF A GAS-PRODUCER

Formation of Explosive Mixtures. All piping and producer doors should be kept tight so that there may be neither leakage of gas out of, nor air into, the producer. When the producer is standing-by, great care should be exercised when making any opening into it, since, while it may apparently have ceased making gas, there may be gas being generated and heat enough present to cause an explosion with the inrush of air through the opening. The producer room should be thoroughly ventilated at all times to avoid danger from gas poisoning. In starting up the producer after a shutdown or stand-by the water should always be turned on in the tower washer after blowing up the fire and before throwing the producer on to the line. This is necessary because, during a shutdown, air may seep into the tower washer, forming an explosive mixture with the gas remaining there, which will be ignited by the hot gases from the producer if they are not cooled by the scrubber water.

Proper Cleaning of Fire. The fire should have sufficient coal, should be clean, and a fire- or clinker-arch should not be permitted to form. In cleaning the fire, care should be taken that the ash is removed from around the walls by poking entirely around the circumference of the lining. If the ash next to the walls is not removed it builds up until it forms a ring which reaches nearly to the top of the fuel bed and offers less resistance to the blast than the coal; more air will consequently travel up around the walls through the ash, in proportion to the cross-sectional area, than through the coal. The coal lying next to the ash ring will be rapidly burned out and the air which does not come into contact with the coal burns with the gas on top of the fire bed and lowers the quality of the gas sent to the engine. If this is allowed to continue the gas will soon become too poor to burn.

Formation of Fire-Arch. A bar should be run down through the center of the fuel bed occasionally to make sure that a fire-arch is not forming. A fire-arch is caused by the coal partially caking and holding up the fuel bed above it, while the ash below it is raked or falls through the grate. This condition can be discovered by running a bar down through the middle, which will meet with great resistance while being driven through the arch and then for the remainder of the distance will meet with no resistance. If it is d

covered in time the arch can be barred down and coal raked from the sides to fill up the hole and the fire soon built up. If it is not discovered, the arch will burn up until it gets so thin that it falls of its own weight, the blast will practically all blow through the center in which there will be very little fuel and the gas will immediately become very lean, causing a shutdown.

Formation of Clinker. If clinker is formed it must be broken up and barred down into the ash bed; if it starts to form and is not attended to, it may clinker entirely across the bed, shutting off the blast. Clinker also assists in the formation of holes through the fire by lessening the resistance to the blast around itself. When clinker is formed in any but a very thick fuel bed, it can be seen on looking down through a poke-hole because of the fact that it is at white heat. If the clinker is located near the wall it will be built up from the wall and fused to the brick and must consequently be broken away from the wall. The first procedure, therefore, is to run the bar with considerable force down at the point of attachment of the clinker to the wall until the clinker is broken loose. It should then be broken up and barred down into the ash bed.

Thin Fuel Bed. If the producer is one in which the fuel bed is carried thin, it must be watched carefully to see that holes are not burned through the fire. If this tends to happen, coal from a thicker part of the bed must be raked over the hole with the bar and then ash in the hole should be barred down.

Red-Hot Bar. In using the bar it should not be kept in the fire more than a few minutes, since it will soon become red-hot and, if the producer is operated under suction, when the bar is pulled out the explosive mixture formed by the air that has leaked into the poke-hole, will be ignited, with possibly disastrous results. The bar should be used for two or three minutes and should then be withdrawn, cooled, and another one substituted.

Circulation in Water Jackets. If any part of the interior of the producer is cooled by a water jacket, great care should be taken that there is no cessation in the flow of the water. If the flow is stopped for any appreciable time, it should not be started again the producer is started after being out of action. If the cold is allowed to strike the hot metal, which would be heated the water ceased to flow, it would crack the metal, would make

a replacement of the part necessary, and would wet the fuel bed, making it difficult to make good gas if the crack were not discovered and the water shut off.

Gas Poisoning. Producer gas, on account of the presence of carbon monoxide, is always poisonous. The carbon-monoxide has a specific toxic effect on the human system, and when inhaled enters into direct combination with the blood. The new compound formed is incapable of carrying oxygen to the tissues and is so stable that it can be broken up only with great difficulty. The action is very insidious, and if the amount that is inhaled is small, the person breathing it may be made almost helpless before he is aware of it. By this time it is often too late for him to escape from the place where the gas is escaping and he becomes unconscious. The symptoms are a sense of discomfort, with throbbing of the blood vessels, severe headaches, the feeling of a tight band around the head and chest causing difficulty in breathing, giddiness, and great debility. In case of poisoning; the first thing to do is to remove the patient to the fresh air and, if he is unconscious, artificial respiration should be applied precisely as in the case of drowning or unconsciousness from electric shock. In handling a patient, and while applying artificial respiration, great care must be exercised to keep the head higher than the lower part of the body. A teaspoonful of aromatic spirits of ammonia in a cup of water should be given to the patient as soon as he begins to show signs of consciousness.

A small bottle of aromatic spirits of ammonia should be kept on hand to use in case of gas poisoning. If the patient is not unconscious and is only suffering from a slight poisoning, the discomforts may be partially relieved by occasionally sniffing the bottle of spirits of ammonia. The discomforts may also be relieved by drinking cold beer or ice-cold milk, although nothing but sleep will relieve the headache and giddiness.

In any event, whisky should never be given the patient, as it tends to fix the carbon monoxide in the system. If the person attempting to attend to a patient, who has been entirely overcome and is unconscious, has had little or no experience with gas-poisoning cases, he should get the patient into the fresh air, send for a physician, and follow these directions as best he may, until the physician arrives.

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